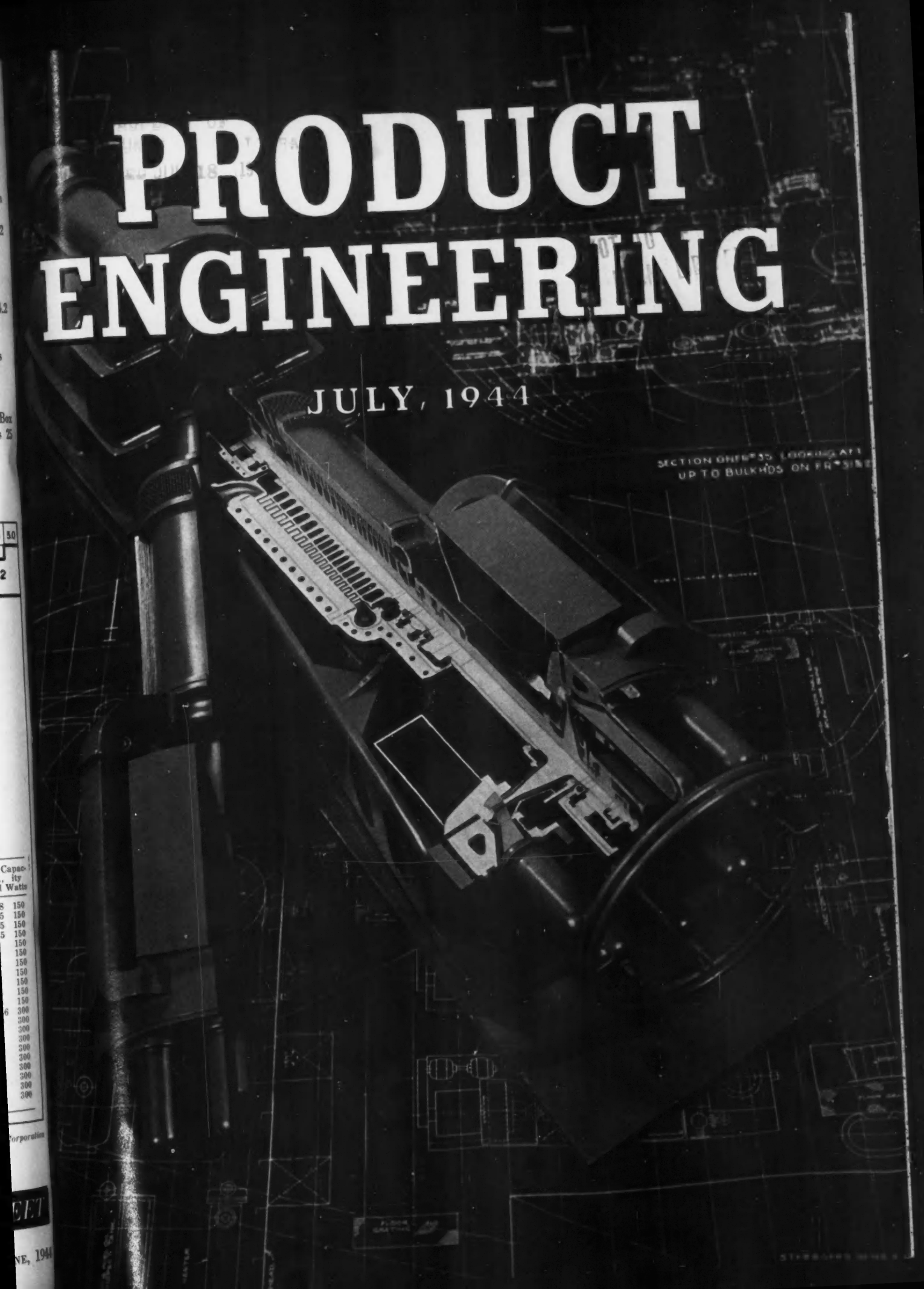
PRODUCT ENGINEERING

JULY, 1944



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to KEEP 'EM IN TOUCH!

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His work, always valuable, now is vital.

No military campaign proceeds without it. The close teamwork between air, ground, and sea arms is possible only through instruments and equipment that keep them in touch though scattered throughout the four quarters of the globe.

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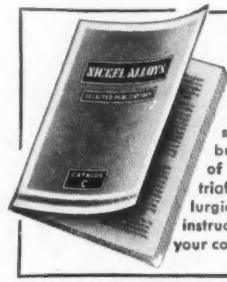

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Subscribers may encounter delay in receiving their copies of PRODUCT ENGINEERING. Such delay is caused by wartime strain on transportation and postal systems, a situation beyond the Publisher's control.

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PRODUCT ENGINEERING. Published monthly. Price 50c per copy. RETURN POSTAGE GUARANTEED. PUBLICATION OFFICES: 99-129 NORTH BROADWAY, ALBANY 1, N. Y. Allow at least ten days for change of address. Communications about subscriptions should be addressed to Director of Circulation, PRODUCT ENGINEERING, 330 West 42nd St., New York 18. Subscription rates: United States, Mexico, Central and South American countries, $5 for one year, $8 for two years, $10 for three years. Canada, $5.50 for one year, $9 for two years, $11 for three years. Great Britain and British Possessions, 36 shillings for one year, 72 shillings for three years. All other countries $6 for one year, $12 for three years. Entered as second-class matter August 21, 1936, at Post Office, Albany, N. Y., under the act of March 3, 1879. Printed in U. S. A. Cable Address: "McGrawhill New York." Member A.B.P. Member A.B.C.

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EDITORIAL AND EXECUTIVE OFFICES: 330 W. 42nd St., NEW YORK 18. Branch Offices: 520 N. Michigan Ave., Chicago 11; 68 Post St., San Francisco 4; 601 W. Fifth St., Los Angeles 14; Aldwych House, Aldwych, London, W. C. 2; Washington; Pittsburgh; Philadelphia; Cleveland; Detroit; St. Louis; Boston; Alanta, Ga.

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IN THIS ISSUE

Cover Picture—Cutaway model of the gas turbine. Kodachrome courtesy General Electric Company.

Industrial Design

Many post-war products will bear the influence of industrial designers. An article on page 433 reviews the results of a survey in which 24 manufacturing companies answered questions regarding their experiences with outside industrial designers. Step-by-step dealings with such outside designers, particularly in relation to engineering departments, are presented.

Thrust Bearing Design By Sleeve Bearing Analogy

Discovery of the laws of fluid friction in bearings has resulted in a number of developments in the field of thrust bearing design. All are based on the inclined slipper principle of producing a fluid film capable of carrying the load. The discussion by H. W. Hamm, see page 437, shows how the hydrodynamic theory for sleeve bearings can be used in design of thrust bearings, and how principles of thrust bearing design can be applied to sleeve bearings.

Interchangeable Assembly

A real interchangeable assembly makes it unnecessary to use such shop tricks and devices as enlarging holes by drill or reamer, substituting smaller

bolts, or selecting matching parts by trial and error. Careful selection on the drafting board of clearances and tolerances on locating dimensions for attaching members permits true interchangeable assembly. Formulas are developed and tables presented for commonest applications in an article starting on page 477 by Frank M. Mallett and Robert H. Lundberg, Curtiss Wright engineers.

Fine Pitch Straight Bevels

On page 473 we present the tentative standards of the A.G.M.A. on fine-pitch straight bevels. Readers who have any suggestions or criticisms concerning these standards are invited to send them in to Mr. Louis D. Martin, Camera Works, Eastman Kodak Company, Rochester, N. Y., who is chairman of the Fine Pitch Gear Committee of the A.G.M.A.

Liquid Honing

Honed finishes are being produced by spraying solutions containing fine abrasives at high velocity. The process was originally intended to improve sand blasting by avoiding dust and inhibiting rust on structural plate. Its applications in deburring, finishing, and similar applications are described on page 468 by A. H. Eppler.

Rubber Mountings

Vibration and shock loads, which prove detrimental to machine operation, can be absorbed through use of many types and forms of rubber mountings suited to a variety of service conditions. On page 482 appear typical mountings available to design engineers.

Graphic Cost Analyses

Full appreciation of the economic advantages to be gained by proper selection of casting method—sand-casting, permanent-mold casting, or die-casting—can only be obtained by picturing the relative cost per piece as quantities increase through the ranges in which production costs for any two methods may be equal. Graphic data based on detailed costs almost automatically selects the method of fabrication. The methods used by Douglas Aircraft Company are described and illustrated in the article starting on page 484.

Fluid-Flow Problems

One of the major steps in many fluid-flow problems, such as predicting the performance of a fluid coupling, involves a trial and error calculation for determining the velocity of flow. R. C. Binder, Purdue University, see page 466, shows how trial and error operations can be eliminated by adding supplemental coordinates to a fluid-flow friction diagram.

Plywood Tubing

Tubing made of plywood is so new that its possibilities have become apparent only during the war. Several uses of plywood tubing that point to peacetime applications were developed in the search for substitutes for critical materials. See page 489 for a review of some of the things that are being done with this unusual material.

Electric Braking Methods—Reversal of Power by Plugging

Demands by industry for economy in production and for safety to both personnel and equipment have made necessary the installation of arrangements for stopping quickly and accurately many electric motor drives. E. H. Hornbarger, control engineer, Westinghouse Electric & Manufacturing Company, see page 449, describes how plugging methods can be applied to a.c. squirrel-cage induction motors, a.c. wound-rotor induction motors, synchronous motors, and d.c. motors. Dynamic braking and regenerative braking will be discussed in future articles.

Practical Fatigue Tests

Since the design of structural members that are to be subjected to fluctuating loads is too uncertain without some knowledge of what fatigue stresses may do to a part, a number of practical test methods were developed by Consolidated Vultee engineers to cause failure of parts by fatigue loading. Criterion of satisfactory design is the number of applications of a suitable load. Choice of the right number and design of testing equipment are described on page 444 by H. O. Boyvey, Chief of Development Laboratories.

Spring-Supported Mountings For Vibration Isolation

Whenever it is necessary to protect the surroundings of a machine from disturbances caused by its operation, engineers instinctively resort to flexibility between the machine and its mounting. M. F. Spotts, Northwestern Technological Institute, see page 441, analyzes a forced vibration system and presents some convenient equations which can be used when proportioning the forces and motions inherent in vibrating machinery. A numerical example shows the application of the method.

Mechanical Properties Of Plastics

Because plastics are synthetics and their characteristics depend on formulation, cure, service conditions and other variables, it is difficult to establish absolute values for mechanical properties. Test data cannot always be relied on since they are often obtained under arbitrary conditions. Hence, designers of plastic parts must be familiar with the various materials so that test data can be properly interpreted and an efficient design created. W. S. Larson, of the General Electric Plastics Department, discusses these factors in the article beginning on page 469, the third in a series covering all phases of plastics part design.

Values For Aircraft AN Steel Bolts

Properties of aircraft AN steel bolts in tension joints, presented in tabular form, see page 503, contributed by R. R. Wiese, administrative engineer, Kaiser Cargo, Inc., Fleetwings Division.

WHAT'S COMING

Effects of Copper Absorbed in Steel

It has long been known that at temperatures in the neighborhood of 600 deg. F., steel will absorb copper and certain other non-ferrous metals. In August PRODUCT ENGINEERING, Mr. F. H. Williams, consulting metallurgist of Canada will show by means of color illustrations, the exact nature of this copper-in-steel absorption and explain how it may result in fatigue failure. It is the possible explanation of failures in brazed parts and shafts or axles that have run hot in bronze bearings.

Types of Plywoods and How Used

A special 16-page insert in August will summarize the types of plywoods, their strength properties and how they are selected and used in the design of products. Data on the high-density impregnated types will be included. Numerous illustrations of typical applications will be shown.

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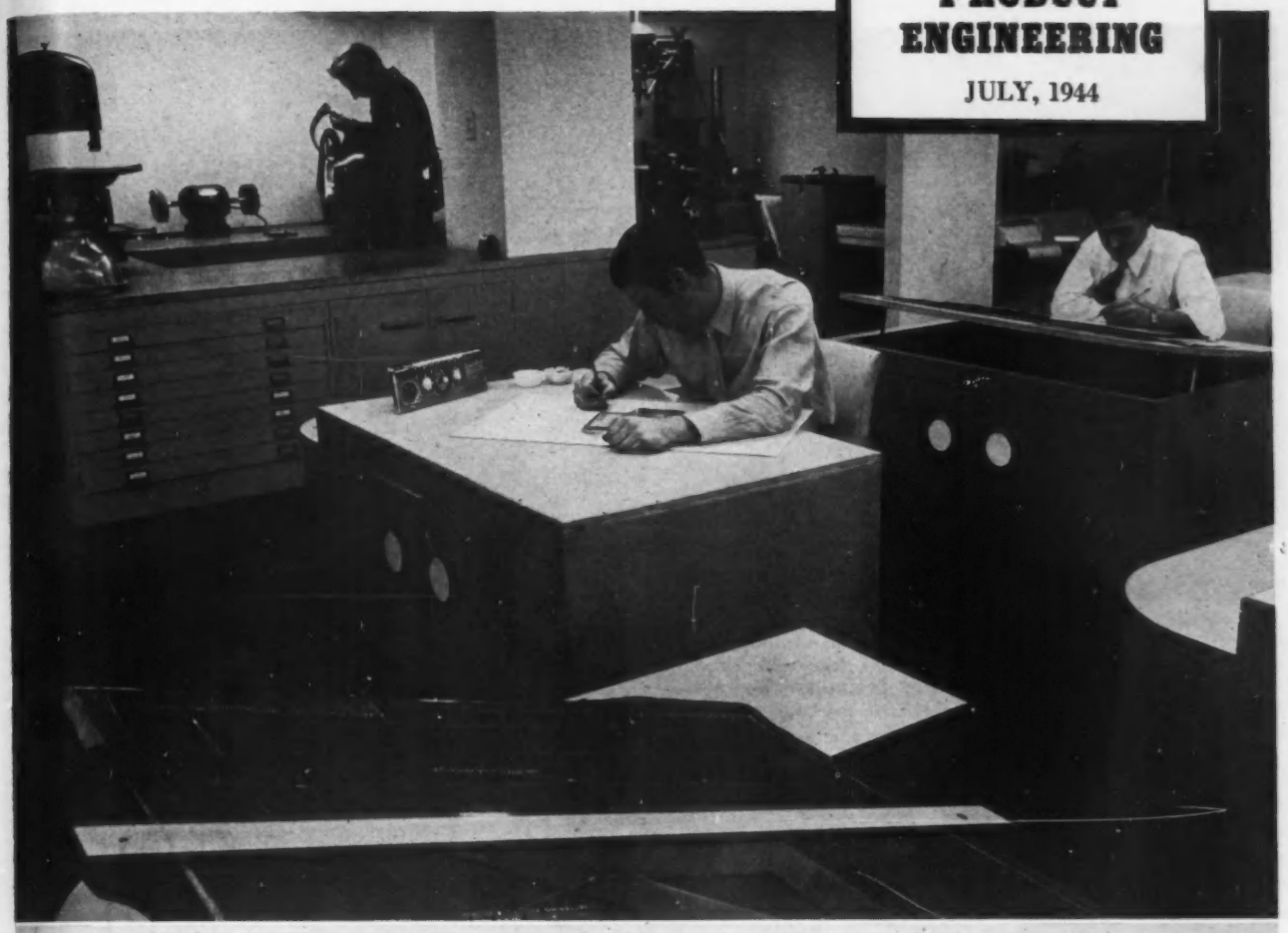


A modern industrial design workshop. Blueprints of a proposed product can be seen in the foreground. Artists are shown at layout tables in the center engaged in sketching design ideas. In the background can be seen part of the equipment used in preparing models. The workshop is that of Dohner & Lippincott.

Industrial Designers Will Influence Many Post-War Products

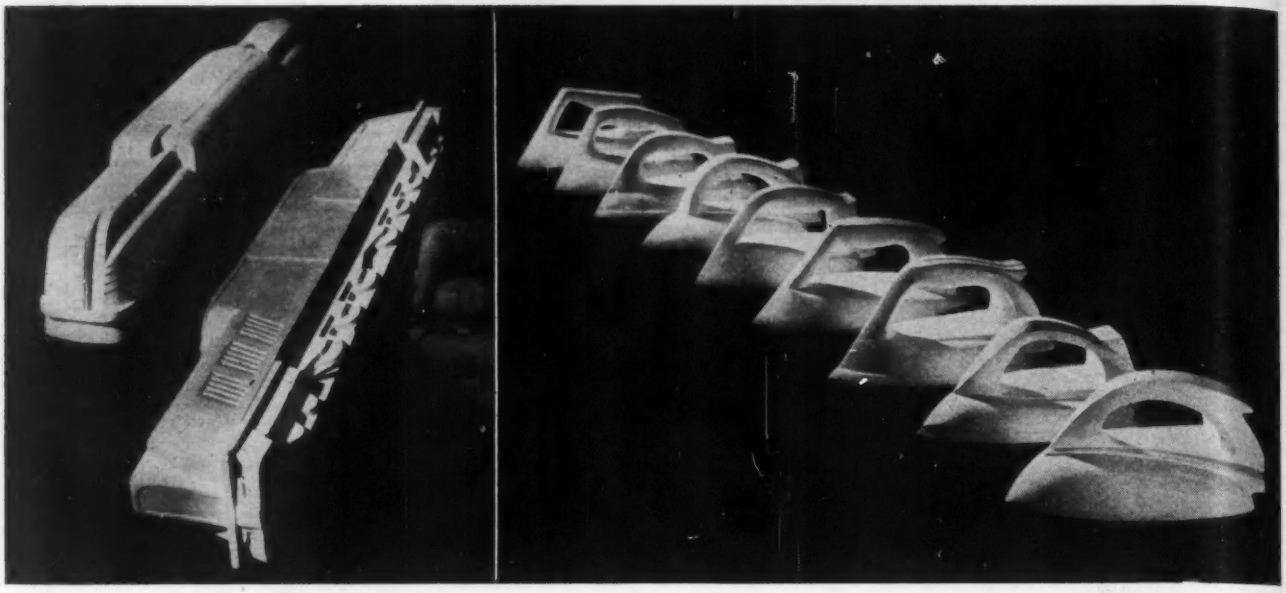
This review combines a survey in which 24 manufacturing executives related their experiences with industrial designers from preliminary negotiations to the finished products. Pertinent questions, such as how to go about getting industrial designers, the services they provide, and how to seek cooperation of plant personnel, are answered.

INDUSTRIAL DESIGN has become firmly entrenched in the American industrial scheme. Services of industrial designers have been reflected in the products of larger companies and will undoubtedly cast their influence on the post-war products of small, medium and large manufacturing firms. Public consciousness in the eye appeal of consumer goods is a leading factor.

The growing importance of art in the design of manufactured products is a logical evolution. In ancient times, manufacturing was done primarily by artists. The relative merits of clay pottery were judged on the basis of artistic values. Each piece of pottery was fashioned skillfully by men whose prime requisite was their artistic ability.

As civilization progressed, the influence and importance of the artist increased. His part in structural projects is demonstrated by design of the pyramids of Egypt. The artist was responsible for the appealing architecture of Greece and Rome. In the Middle Ages, the artist's influence extended to the design of furniture, glassware, pottery, paintings and jewelry.

Real progress in industrial design



Dohner & Lippincott

Plaster models are an important tool of industrial designers for visual presentation of products. Proposed locomotive designs are shown being prepared in plaster to demonstrate appearance factors. Nine plaster models of an electric iron, each having a different handle, have been prepared by the designer as the basis of a conference at the client's plant. Various department executives—engineering, sales and production—will examine the models and consumer research reports will be studied to determine which one will best satisfy the average housewife. This ties in with the marketability of contemplated products.

stems from the textile field. Rugs were woven in private homes until the French established state-owned factories centuries ago. Artists were then harnessed into the industrial picture to design rug patterns.

German schools started teaching industrial phases of design as it applied to textiles and household goods. Later similar schools were opened in France and England.

With the advent of modern machinery, mechanical goods and the mass-production system, the artist's influence ceased to expand and he descended from his prominent position. This was an understandable change. The initial problems of the industrial age were technical rather than artistic. Although early models of machinery reflected the influence of art as shown by pictures of ornate lathes and engines, the increasing pressure to perfect design and production squeezed the artist out of the picture for a time.

Tremendous progress in engineering design and mass production marked the World War I period. Many basic engineering problems were solved and fundamentals of mass production became firmly established. Soon after the close of World War I the public became more critical of the products of industry. Technology made available vast quantities of new machines, perfected to a degree that left little to be desired from the standpoint of serviceability, quality and durability. Automobiles, vacuum sweepers, washing machines, radios and other mechanical equipment for the home became obtainable at relatively low prices in the Twenties.

Satisfied with the price and service qualities of mass-produced items, consumers began casting a critical eye at the appearance of products. They demanded that the product have an attractive appearance to provide them with "pride of ownership." Gradually this demand spread to other products, but it was not until the advent of the industrial designer in the middle Twenties that industrialists became thoroughly impressed with the public's insistence that products possess eye appeal because they "had to be lived with" and were a part of the consumer's daily environment.

In seeking to extend markets for products, advertising men are credited with initiating market research. They realized that to fulfill their objective steps were necessary to determine product appeal from the consumers' standpoint. It became evident that there was no close relationship between the product the manufacturer turned out and the market. Furthermore, manufacturers were not benefiting from consumer experiences. It focused attention on the importance of market research, and indicated that products had to go beyond functional advantages. Thus, the importance of aesthetic appearance in products became widely recognized by industry in the Thirties.

Because few manufacturers had the necessary artistic talent within their organization, it became necessary for them to procure the services of industrial designers to redesign their products for sales appeal in order to compete with foresighted leaders. It was this obvious need that brought into being the industrial designer as we know him today. Although his services were obtained primarily for the purpose of improving the appearance of the product, the modern industrial designer was compelled to extend his influence to render assistance in the selection of materials and manufacturing processes to make the product serviceable, durable and attractive. The modern industrial designer also may make market studies in order to pre-establish the consumer tastes and desires. In fact, in many instances this becomes a most essential part of his services.

In a recent speech at the annual meeting of Associated Industries of New York State, Inc., Harold Van Doren, prominent industrial designer, defined his profession as follows: "The modern industrial designer is concerned with 3-dimensional products, equipment and machines made only by production methods, as distinguished from traditional handcraft methods. His aim is to enhance the desirability of these products by: (1) Increasing convenience and improving adaptability of form to function; (2) attracting buyers by applying a shrewd knowledge of consumer psychology; and (3) employing to the fullest the aesthetic appeal of form, color and texture."

Because industrial design is a young profession, there are relatively few qualified in the profession. This situation is now creating considerable difficulty for innumerable manufacturers who suddenly realize that they must

procure the services of such designers in order to retain their position in the competitive market.

The difficulty is further intensified by the fact that there is no definite source of supply for experienced industrial designers, although several general procedures are open to the manufacturer who is seeking the services of such qualified persons. A number of industrial publications describe products that have been designed by industrial designers. Occasionally one will find articles in magazines describing the work of various industrial designers or articles written by industrial designers discussing some phase of their professional activities.

A list of the names of industrial designers associated with the American Designers Institute can be procured from the organization. The Society of Industrial Designers is being formed and has enrolled 14 of the top designers in the country. All of these leads, however, result in most instances in a mere list of names which gives no indication of the prominence, abilities or fees of the designer. Manufacturers cannot narrow the field to designers fitting their particular needs.

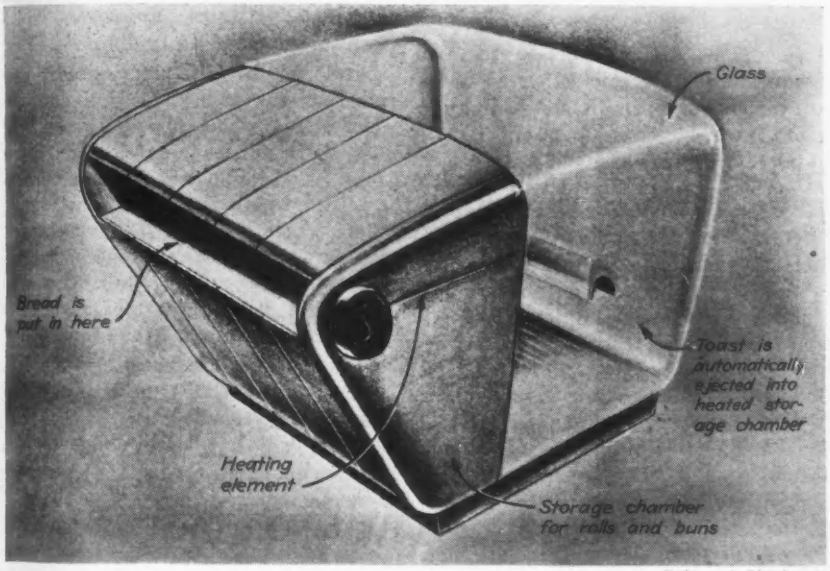
Perhaps the most effective way of obtaining names of industrial designers who might meet the needs of a given manufacturer is to make inquiries among other manufacturers who have availed themselves of the services of industrial designers. In some instances a manufacturing company will advertise that its products have been designed by a certain designer.

The President or Vice President of a company usually negotiates the hiring of an outside industrial designer or a product development consultant who cooperates in disclosing the marketing needs. Before engaging an industrial designer, it is wise for company officials to confer with their chief engineer to sell him on the reasons for hiring an industrial designer. This preliminary step is extremely important because without enthusiastic cooperation between the industrial designer and the chief engineer, little can be accomplished, or what is achieved generally proves expensive and less effective. This does not imply that the chief engineer should select the designer and do the hiring. A survey of 24 companies that engaged industrial designers showed that in all but two instances the hiring was performed by the President or Vice President of the company.

Basis for compensation is an important consideration in negotiating with an industrial designer. The usual and most workable basis appears to be on a monthly retainer plus expenses. Expenses including traveling, cost of models, pictures and drafting work. Product development consultants include the cost of product and market surveys. This basis seems to have a protective angle in that the manufacturer is free to dispense with the designer's services at any time by giving, usually, 30 days' notice of his intentions. Ethical designers do not handle competing accounts.

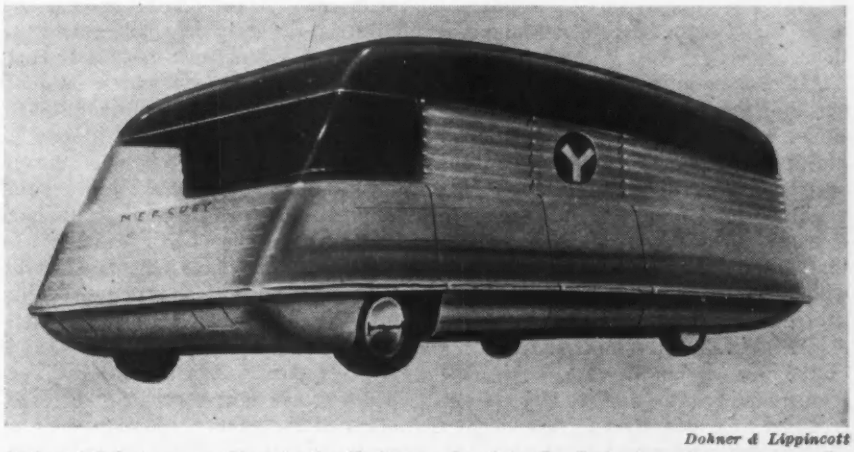
Sometimes the designer is paid a lump sum for his work. Not one of the 24 manufacturers covered in the survey paid the designer on a royalty basis such as a percentage of the dollar sales of the designed product. Such a basis of compensation is frowned upon by most of the recognized industrial designers. This basis is confined mostly to inventions. One firm representative wrote that in his field "a per diem basis is the only practical method of compensation in our opinion because we have to do everything ourselves except that we use them (industrial designers) to aid us in eye appeal." Most designers however, go beyond the eye appeal phase and recast the entire product. Ability to do this is considered an important qualification.

After the industrial designer has been hired, considerable thought should be given to his introduction to the plant personnel and familiarizing him with plant activities and equipment. It is considered good policy to have the chief engineer introduce the designer to the engineering department personnel. This tends to establish the industrial


Glass
Bread is put in here
Toast is automatically ejected into heated storage chamber
Heating element
Storage chamber for rolls and buns

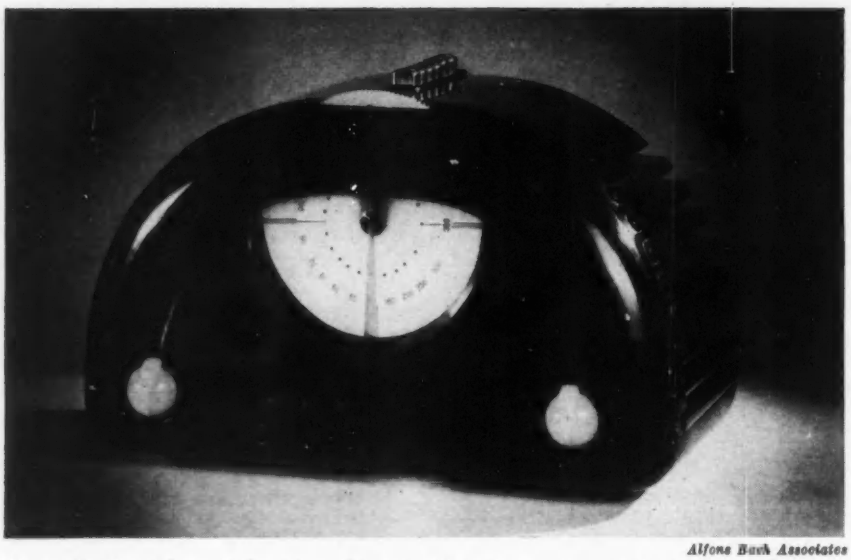
Dohner & Lippincott

Novel yet practical ideas are incorporated into designs such as this automatic storage toaster. When the bread is toasted, it is ejected and piled in a storage chamber where it is kept warm until wanted. A housewife's suggestion might lie behind this design.



Dohner & Lippincott

Industrial designers will undoubtedly have a hand in the designing of post-war trucks. The one sketched is one designer's conception of a useful delivery truck. Engineering requirements and numerous other factors, such as materials and safety, must be considered by the designer. Note the elimination of windshield posts for better vision.



Alfons Bach Associates

Streamlined, table model radio cabinet typifies the appearance factor created by an industrial designer. Engineering considerations are important in such designs.

designer in the eyes of the engineering department members.

No company should expect the industrial designer to take off his coat, roll up his sleeves, and in a few minutes produce a wonderful redesign of the complete line of products. It must be recognized that the industrial designer requires time to make a case study of the products and acquaint himself with manufacturing equipment available in the plant.

Many prospective products will make it necessary that the industrial designer carefully survey consumer tastes and desires. The industrial designer often will have to extend his activities to cover market studies and research. A manufacturer cannot afford to guess the extent of acceptance of his product. He should be able to forecast it. Some designers, highly skilled in their own fields, may not have market research facilities but there are special groups including advertising agencies that can provide this type of service to supplement the designers' personal observations.

After acquainting himself with the needs of the project, the industrial designer usually makes sketches of possible new designs. These pictorial sketches are only preliminary. It might be fatal for the manufacturer to consider them as anything more than that. Gradually, after consultations with the chief engineer and with ideas presented by various members of the engineering staff, the form and features of the new design develop. The new features may not be and usually are not confined to appearance factors. Many of the jobs performed by industrial designers have resulted in new service values or improved operating characteristics in the new product. Such features might include latches that are simple to operate, doors that are more convenient to open, handles that are more accessible or easier to operate. Or it might be some additional feature such as a headlight on a vacuum sweeper or a light inside a refrigerator.

Obviously the industrial designer cannot be expected to be an expert on all phases of engineering design and production problems in addition to being a specialist in the creative work of incorporating sales appeal in a product. But industrial designers are or should be well posted on general engineering and production principles, materials of construction and shop operations even though their knowledge of certain specialized technical problems of a given factory might be slight. There are some types of products, of course, which do not require engineering, such as household bric-a-brac.

"Industrial design as applied to aircraft," commented one executive, "involves many compromises in that the industrial designer does not control the entire design and must constantly compromise with aerodynamics, weight, space and unusual service conditions."

This emphasizes the importance of enthusiastic cooperation between the engineering department and the industrial designer. Each can contribute tremendously to the efforts of the other. In this connection, managements must be prepared to stifle clashes of personality and pave the way for teamwork. Oftentimes it resolves itself into a "give-and-take" proposition. Only by such cooperation can best results be achieved.

One of the greatest contributions of the successful industrial designer, perhaps, is his ability to create new viewpoints in the engineering department and to stimulate the men in the engineering department. Seven out of 24 chief engineers reported that the industrial designer was responsible for creating a spirit of enthusiasm among the engineering department personnel. Old and young men got into the spirit of the "new adventure" and vied with each other in bringing forth new ideas that might be incorporated in the product being redesigned.

One chief engineer described how the industrial designer took pains to see to it that the men in the engineering department received full credit for all the ideas they presented. In many instances, an engineering staff member initiated a new idea and when the seed developed and brought forth results, the industrial designer saw to it that the engineer received recognition.

Another firm executive stated, "Our whole organization, sales (field), production as well as engineering, was stimulated. Helps overcome inhibitions and knocks out tradition."

It is emphasized that in every case wherein an industrial designer was successful he had consulted freely with the chief engineer in development of the product. In 23 out of 24 companies investigated, all but one reported that there was a good spirit of cooperation between the industrial designer and the chief engineer and his staff. The only exception stated that the cooperation was "fair and might have been better."

One wonders to what extent the final design as created by the industrial designer is changed later by the engineering department. In about one-half of the instances reported, an appreciable number of changes were made in the design submitted by the industrial designer. In 85 percent of the cases, however, these changes were made during development of the design and in cooperation with the chief engineer. In only half of the instances were the changes extensive. Usually the changes were made primarily for the purpose of speeding production.

In 80 percent of the instances, the product was definitely a commercial success. In one instance, the manufacturer reported that the product had not been on the market long enough to determine the success of it. Of greater significance is the fact that all of the manufacturers contacted reported that they were satisfied with the services rendered by the industrial designer and they all considered that the money had been well spent. In all but one instance, manufacturers reported that the money spent proved a profitable investment.

The importance of securing increased marketability through design cannot be underestimated. It is fundamental to consumer and service goods.

Thrust Bearing Design By Sleeve Bearing Analogy

H. W. HAMM

Determination of allowable bearing pressure, the coefficient of friction and the slipper angle for inclined slipper bearings by a method based on the hydro-dynamic theory for sleeve bearings. Conversely, application of slipper bearing design principles to sleeve bearings indicates that allowable sleeve bearing pressures are high for slow-speed shafts.

WHEN confronted with the problem of finding a suitable bearing to take up a thrust load, the design engineer must satisfy the following requirements to produce an economical design. Unit bearing pressure must be a maximum as dictated by the characteristics of the bearing material and the hydro-dynamic conditions. The coefficient of friction must be held at a minimum figure for safe fluid friction operation, thereby eliminating the danger of boundary lubrication with metal to metal contact except at starting and stopping.

The life of the bearing is also of importance. A bearing operated under fluid friction conditions has a long life particularly at high speed. As long as the bearing is supplied with clean oil of the proper viscosity it is practically indestructible.

The only thrust bearing operating under fluid friction condition is the inclined slipper bearing which consists of a flat rotating disk and a number of stationary wedge shaped bearing blocks. Between these the wedge shaped load carrying oil film is formed. Some types of inclined slipper thrust bearings, by flexibility or hinges, adjust themselves automatically to the most favorable friction conditions. There is no reason to assume that solid slipper bearings cannot be calculated and designed for the same maximum efficiency by determining in advance the correct angle of incline for the given speed, diameter, load and oil viscosity.

Sleeve bearings have been tested thoroughly. A definite relationship has been established betwen the bearing constant Zn/p and the coefficient of friction. The factors in the bearing constant are

Z = absolute viscosity in centipoises
n = r.p.m. of shaft
p = bearing pressure, lb. per sq. in.

It seems advantageous to use this relationship as a basis for thrust bearing design also. Fundamentally there is no difference in the operation of the two bearing types.

For the inclined slipper bearing the following equation has been established (see "Fluid Mechanics"—Dodge and Thompson, 1937 Edition, page 464) for the pressure rise resulting from the pumping action of a flat plate moving in the direction indicated in Fig. 1. With

x = distance from origin
p_x = pressure rise above atmospheric pressure at distance x from origin
Z = absolute viscosity of oil
v = velocity of bearing plate
L = length of slipper
α = angle of incline in radians
A = distance between origin and intersection of slipper with moving plate

$$p_x = \frac{6Z\,v\,x\,(L-x)}{\alpha^2\,(2A-L)\,(A-x)^2}$$

The pressure distribution along the length of the slipper for different L/A values is shown in Fig. 2. For a ratio L/A equal to 1.0 the pressure rise is infinite at x equal to L because the distance between slipper and plate has become zero. For L/A less than or equal to 0.7 the curves for the pressure rise are of such a shape that the average value of the pressure is approximately

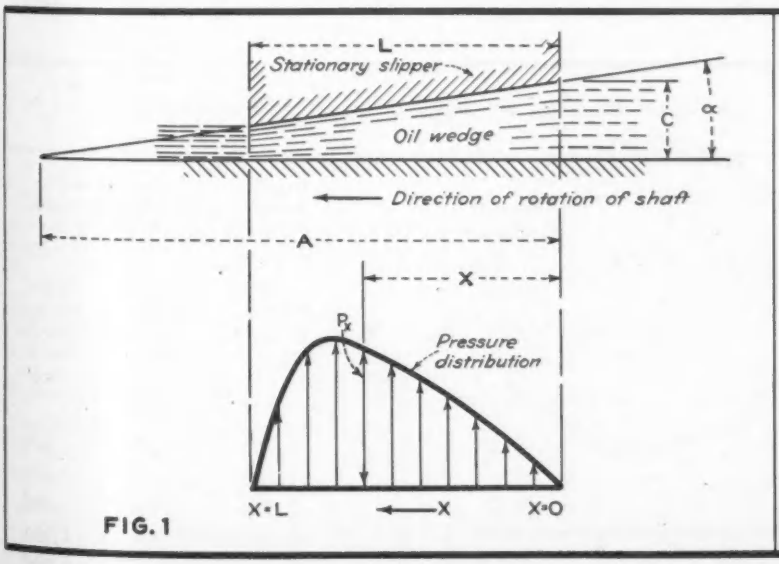


FIG. 1

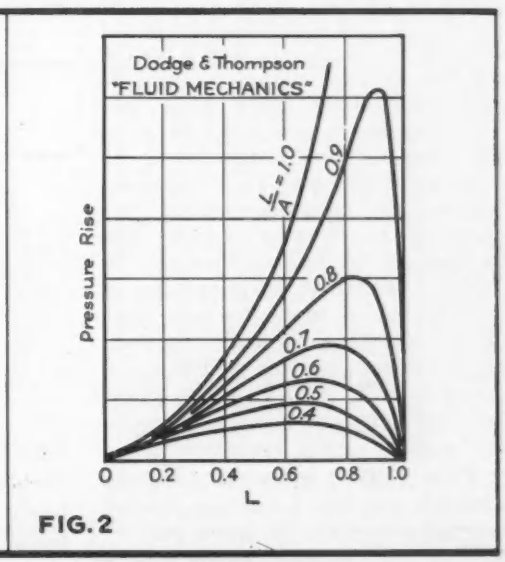


FIG. 2

Fig. 1—Pressure distribution in wedge-shaped oil film in a plane thrust bearing.
Fig. 2—Oil pressure distribution on inclined slipper for different L/A values.

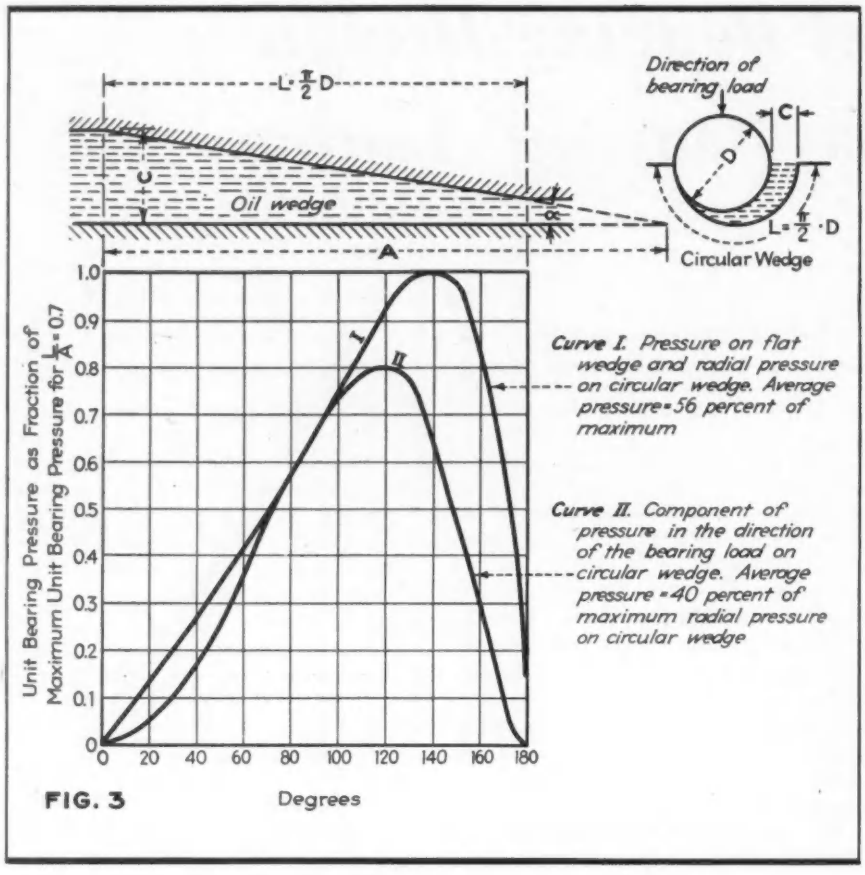


Fig. 3—Analogy between pressure distribution for inclined slipper and fluid film sleeve bearing.

0.56 of the maximum. This fraction multiplied by the elastic limit of the bearing material gives the maximum allowable average pressure under which a bearing can operate safely without affecting the bearing material.

For a sleeve bearing, which may be considered an inclined slipper bearing wrapped around a shaft, only the component of pressure in the direction of the bearing load is effective in carrying the load. In Fig. 3, this force is plotted and the average component over the total length of the film is given. For L/A less than or equal to 0.7 this average value is approximately 0.4 of the maximum radial pressure. For sleeve bearings the allowable average unit pressure is therefore 0.4 times the elastic limit of the bearing material. In Table 1, the elastic limits and the allowable bearing pressures are given for a few bearing materials. For babbitt the values given are valid at a temperature of 212 deg. F., which allows for varying factor of safety for operation at lower oil-film temperatures.

While Table 1 gives the maximum allowable pressures for a given bearing material regardless of speed and oil viscosity, the bearing constant Zn/p determines the pressure which can be used with certain hydro-dynamic conditions. Whichever value is the smallest is the one which must be used in the design of the bearing. For sleeve bearings the relationship between oil viscosity, speed, pressure and coefficient of friction is known. (See "Bearing Design by Hydro-Dynamic Theory", H. W. Hamm, PRODUCT ENGINEERING, page 422, July 1943). These same values can be applied to the inclined slipper bearing by wrapping the slipper around a semi-circular area and using the diameter of the full circle as the equivalent diameter D_s of the slipper as shown in Fig. 3. The arrangement of the slippers is shown in Fig. 4, in which R_m is the radius of the center of pressure.

$$R_m = \frac{2\,(R^3 - r^3)}{3\,(R^2 - r^2)}$$

With the radii expressed in feet, and n is the r.p.m. of shaft, the average velocity in ft. per min. of the flat plate in relation to the inclined slipper is

$$v = 2\,\pi\,R_m\,n$$

With m equal to the number of slippers

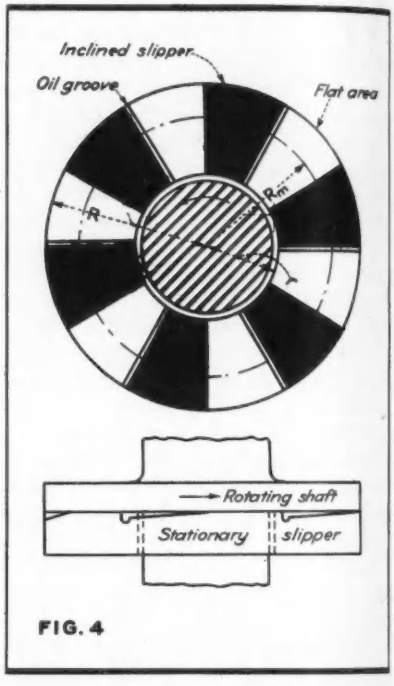


Fig. 4—Arrangement of stationary slipper in a thrust bearing for a rotating shaft.

Table I—Bearing Pressures for Fluid-Film Bearings

Material	Material Specifications					Elastic Limit, lb. per sq. in.	Max. allowable average pressure, lb. per sq. in.	
	Cu	Sn	Sb	Pb	Ni	(For Babbitt, at 212 deg. F.)	Sleeve bearing, 0.4×El. Limit	Thrust bearing, 0.56×El. Limit
Babbitt.....	8.3	83.3	8.3	....	....	1,300	520	730
Babbitt.....	3.0	75.0	12.0	10.0	....	1,550	620	870
Babbitt.....	....	10.0	15.0	75.0	....	1,350	540	750
Bronze.....	80	10	....	10	....	9,000	3,600	4,900
Bronze.....	84	10	....	2.5	3.5	18,000	7,200	9,800

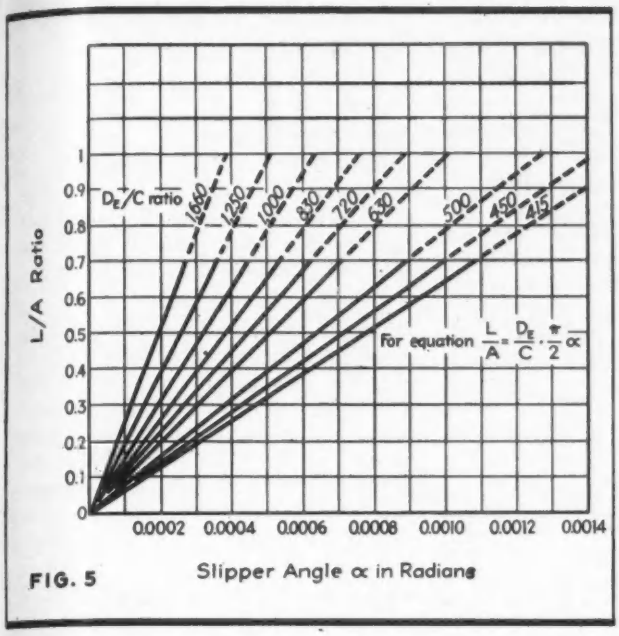


FIG. 5

Fig. 5—Relationship between slipper angle and the L/A ratio for different equivalent diameter to clearance ratios.

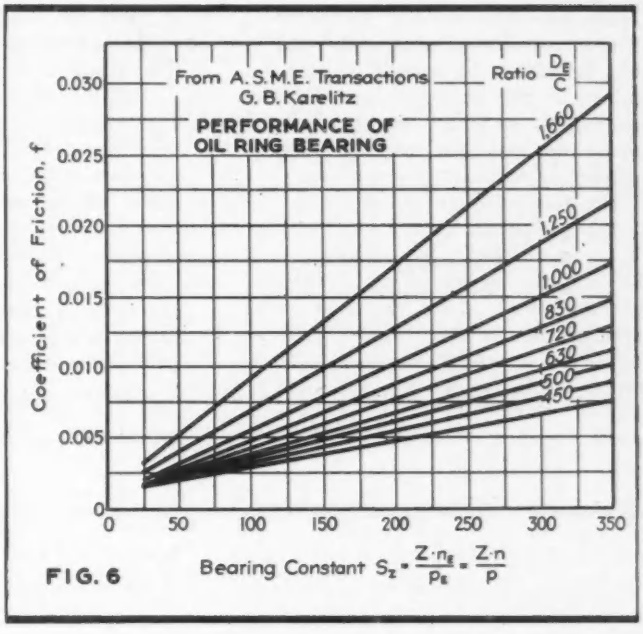


FIG. 6

Fig. 6—Equivalent diameter to clearance ratio to fulfill conditions imposed by coefficients of friction and bearing constants.

(with flat spaces of equal size and number), the length L of the slipper at radius R_m is

$$L = \pi R_m/m$$

The diameter D_E of the equivalent sleeve bearing is

$$D_E = 2L/\pi = 2R_m/m$$

The number of revolutions N_E of the equivalent sleeve bearing is

$$N_E = \frac{x}{\pi D_E} = \frac{v\,m}{2\pi R_m}$$

This value of N_E can be used in the same manner as n is used for the sleeve bearing, to determine the unit bearing pressure with a given bearing constant Zn/p greater than or equal to 36.

To complete the design, the angle α of the slipper and the equivalent diameter to clearance ratio D_E/C remain to be determined. From Fig. 3 it can be shown that

$$\frac{L}{A} = \frac{\pi D_E \tan\alpha}{2C}$$

where because of the small magnitude of the angles used, the angle α may be substituted for tan α, the equation then becomes

$$\frac{L}{A} = \frac{\pi D_E \alpha}{2C}$$

Angle α is a function of L/A and D_E/C. This relationship is plotted in Fig. 5 for a series of D_E/C ratios for which the coefficient of friction has been determined experimentally in sleeve bearings.

The D_E/C ratio can be selected from Fig. 6 to give a satisfactory coefficient of friction for a given bearing constant Zn/p. For selected ratios of L/A and D_E/C the slipper angle α can be found from Fig. 5. This angle is very small, and to obtain a slipper which can be accurately machined it may sometimes be necessary to select a D_E/C ratio which results in a coefficient of friction greater than the theoretically desirable value.

Sleeve Bearings

In the preceding discussion the hydro-dynamic theory for sleeve bearings was applied to axial thrust bearings of the inclined slipper type. Inversely the experience gained in the operation of the latter can be applied to the sleeve bearing, which is a variation of the slipper bearing the length of which is equal to one half of the shaft circumference in a sleeve bearing. The bearing constant Zn/p determines the allowable pressure. If it is possible, as has been done with the thrust bearing, to divide the load-carrying area into a number of small slippers, the equivalent speed for each slipper increases with the number of load-carrying slippers. Then with Zn/p constant and with viscosity Z constant, and

n = r.p.m. of shaft in plain sleeve bearing
N_E = equivalent r.p.m. of multiple slipper bearing
p = bearing pressure in plain sleeve bearing
p_E = bearing pressure in multiple slipper bearing
m = number of slippers carrying load
$N_E = mn$

and hence $p_E = mp$

That this deduction is correct has been proved by Mr. G. Fast in his tests of multiple film sleeve bearings. (See "The Flexible-Sleeve Multiple Oil-Film Radial Bearing," Gustave Fast, A.S.M.E. *Transactions*, Vol. 63, No. 8, November, 1941). The main characteristic of these bearings is the multiple oil film which is produced by dividing the sleeve into a number of slippers separated by suitable grooves as shown in Fig. 7. These grooves must be wide and deep enough to cause a pressure drop to atmospheric as the oil is discharged from the narrow end of the wedge. In the grooves the oil mixes with the cooler oil from the outside, and the mixture enters the next wedge.

In addition to dividing the bearing sleeve into a number of wedges, Mr. Fast also made these wedges flexible, assuming that they would then adjust

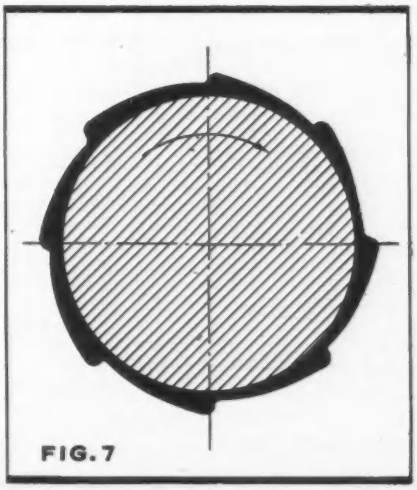


FIG. 7

Fig. 7—Bearing with multiple oil film produced by dividing the sleeve into a number of slippers and deep grooves.

themselves automatically to the most efficient conditions. But even without this flexibility, by merely cutting oil grooves in the sleeve, it is possible, as can readily be seen from Fig. 8, to increase the unit pressure by using a number of load-carrying wedges.

The number of wedges shown in Fig. 8 equals one-half of the total number of wedges around the circumference. The angle of incline for all the wedges is the same, while the L/A ratio changes. At the narrow end L_1/A_1 is large which results in a large D_E/C ratio. While toward the wide end of the oil wedge the value of L_1/A is smaller, with a consequent decrease in D_E/C and the coefficient of friction. The total friction is the mean of that of the individual wedges. This figure is only slightly higher than the coefficient of friction obtained in the conventional sleeve bearing.

As proof of the validity of these deductions which lead to a higher unit load of low-speed fluid film bearings by application of the multiple oil-film principle, the following example may serve:

Shaft diameter, $D = 3$ in.
Effective bearing width, $l = 0.75$ in.
Oil-film temperature = 100 deg. F.
Oil viscosity, $Z = 185$ centipoises for S.A.E. No. 50 oil
Bearing Constant, $Zn/p = 40$

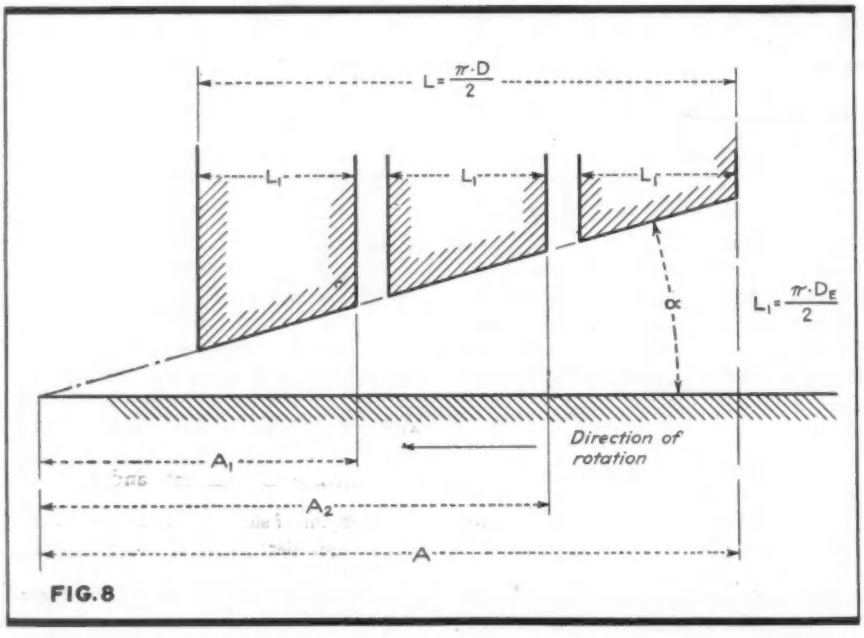


FIG. 8

Fig. 8—Unit bearing pressure is increased by using a number of load-carrying-wedges formed by cutting oil grooves in bearing sleeve. Total number of wedges equals six.

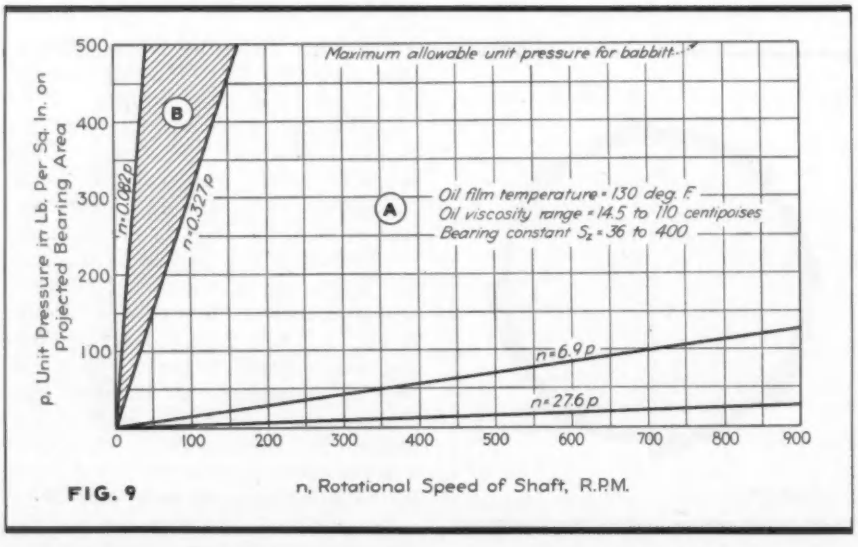


FIG. 9

Fig. 9—Comparison of fields of application for multiple oil-film bearing and conventional sleeve bearings. Area "A" is range of conventional sleeve bearing. Area "B" is additional range gained by multiple oil-film bearing in the low speed field.

Revolutions of shaft per minute, $n = 50$

Allowable bearing pressure

$$p = \frac{Zn}{40} = \frac{185 \times 50}{40} = 231 \text{ lb. per sq. in.}$$

Total allowable load

$$P = pDl = 231 \times 3 \times 0.75 = 520 \text{ lb.}$$

This is the allowable load for the conventional sleeve bearing. For a multiple film bearing with a total of eight slippers the allowable load will be

$$P_E = 4 \times 520 = 2{,}080 \text{ lb.}$$

which is the allowable load given by Mr. Fast for this type and size of bearing. The average unit pressure for this case is 924 lb. per sq.in. and the maximum unit pressure is 924/0.4, or 2,310 lb. per sq.in., which is still below the elastic limit of babbitt at a temperture of 100 deg. Fahrenheit.

The advantage of the multiple oil-film bearing in the low-speed field is shown diagrammatically in Fig. 9. It also shows that at some speeds this bearing overlaps the range of the conventional sleeve bearing. This range has been found by using the formula for the bearing constant S_z equals Zn/p and introducing the minimum and maximum allowable S_z as determined by stability of fluid friction conditions and by friction.

Minimum $S_z = 36$, this value includes a safety factor for stability
Maximum $S_z = 400$, at this value and a D/C ratio of 1,000, the coefficient of friction f equals 0.02, which is high

By introducing the maximum and minimum viscosity of commercially available oils for the normal oil-film temperature of 130 deg. F.

Minimum $Z = 14.5$ centipoises
Maximum $Z = 110$ centipoises

The range of rotational speed n as a function of the unit bearing pressure on the projected area p can be obtained from

$$n_{min} = 0.327\ p$$
$$n_{max} = 27.6\ p$$

For the multiple oil-film bearing with eight slippers used in the example, the minimum speed will be

$$n_{min}/4 = 0.327p/4 = 0.082\ p$$

The maximum speed, 6.9 p, comes within the range of the conventional bearing and is for this reason without practical value.

For most operating conditions, as shown in Fig. 9, the maximum allowable bearing pressure p is a constant which is determined by the material of the bearing sleeve. From Table I, a safe figure for babbitt seems to be 500 lb. per sq.in. In cases where room temperatures vary and the bearing loads cannot be determined accurately, a safety factor will have to be included to take care of these and other operating contingencies.

Spring-Supported Mountings For Vibration Isolation

MERHYLE F. SPOTTS
Northwestern Technological Institute

Explaining how the general equation for forced vibration without damping can be expressed in convenient forms for use in designing spring-supported mounts for reducing the forces and motions of vibrating machinery. Curves are included for the graphical solution of the equation for any desired percentage of isolation.

UNLESS carefully guarded against, periodic forces exerted by machines on their mountings at regular time intervals set up resonant vibrations out of all proportion to the magnitude of the originating disturbance.

The common causes of periodic forces in machines are unbalance of rotating parts, inertia forces of reciprocating parts, and variable torque reactions.

In making an analysis of the forces and displacements in a vibrating system, the diagrammatic form, shown in Fig. 1, of the forced vibration system has been somewhat idealized.

The rotor, frame and base, shown in Fig. 1, have a total weight of W lb. supported by the spring of constant k lb. per in. of deflection. It is assumed that the spring follows Hooke's law for proportionality between load and deflection. Since only vertical motion will be considered the weight is constrained to move in this direction only by means of frictionless rollers at the sides. When the rotor is at rest its center is on the line of static equilibrium which accordingly is chosen as the origin for the measurement of the displacement x. With the rotor turning, the equivalent of an eccentric weight W_o produces a centrifugal force F_c. With the rotor revolving at an angular velocity of ω radians per sec., at any instant t the body will have a displacement x and a corresponding vertical disturbing force $F_c \cos \omega t$, since angle ωt indicates the instantaneous location of F_c. Should the rotor be stopped and the system undergo free vibration vertically, the frequency would be ω_n radians per sec., in

$$\omega_n^2 = kg/W \qquad (1)$$

where g is the gravitation constant of 386 in. per sec. per second.

BASIC EQUATION. The derivation of the basic equation for forced vibration without damping will not be repeated here since it can be found in many books on mechanics and vibration theory. This equation is

$$x = \frac{F_c}{k} \frac{1}{1 - \frac{\omega^2}{\omega_n^2}} \cos \omega t$$

$$= \frac{F_c}{k} \frac{1}{1 - \frac{\omega^2 W}{kg}} \cos \omega t \qquad (2)$$

It is assumed that whatever friction or damping forces may be present are small enough to be neglected. As shown in Fig. 1 the reaction at the base of the spring consists of the weight W plus the force kx caused by the deflection of the spring. The inertia force Wa/g is used in the derivation of Equation (2), but is of no further concern. During one-half of the cycle the body is located above the line of static equilibrium and all forces except W in Fig. 1 will be reversed. Equation (2) is, of course, valid for the entire cycle.

When the situation is as shown in Fig. 1 the operation is taking place "below resonance," that is, whenever ω_n is greater than ω. For such operation there is no isolation as an examination of Equation (2) indicates that x and consequently the reaction force kx have considerable magnitude. The presence of the spring makes the situation worse than if W were bolted down solidly to the foundation. The force in the latter case then would be merely F_c without magnification.

Should ω_n and ω have values very close to each other, amplitudes of great magnitude may be built up. Operation then would be said to be taking place "at resonance." At this time the damping, even though slight, will have a large effect on x and therefore Equation (2) is no longer applicable.

In order to isolate the vibrating system from the foundation it is necessary that ω_n in Equation (2) be considerably smaller than ω. Operation then will be "above resonance." The various forces will be related to each other as shown in Fig. 2, which shows the body at the maximum upward displacement x_m. When the disturbing force F_c and displacement x_m are 180 deg. out of phase, the term $\cos \omega t$ and the expression for x_m in Equation (2) become

$$\cos \omega t = -1$$

$$x_m = -\frac{F_c}{k} \frac{1}{1 - \frac{\omega^2 W}{kg}} \qquad (3)$$

REDUCING THE AMPLITUDE. The unbalanced force F_c usually depends upon the

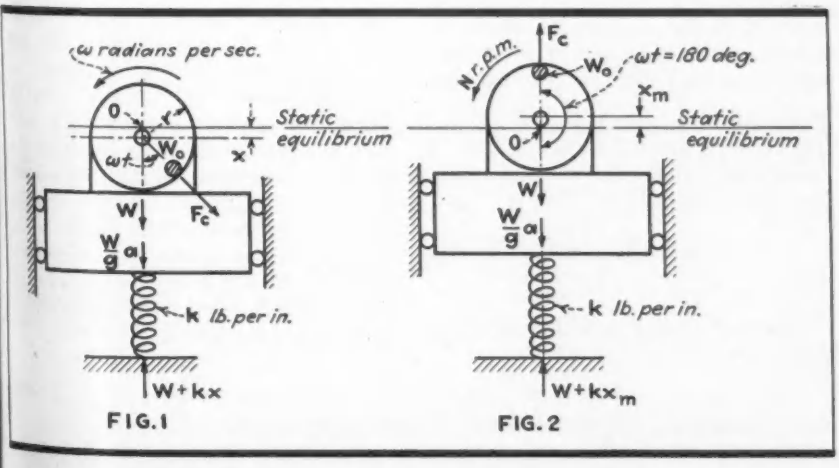


Fig. 1—Spring-supported rotor, frame and base acted upon by a harmonic disturbing force. Conditions indicated are for operation below resonance. Fig. 2—Forces acting in a spring-supported system when operation is above resonance.

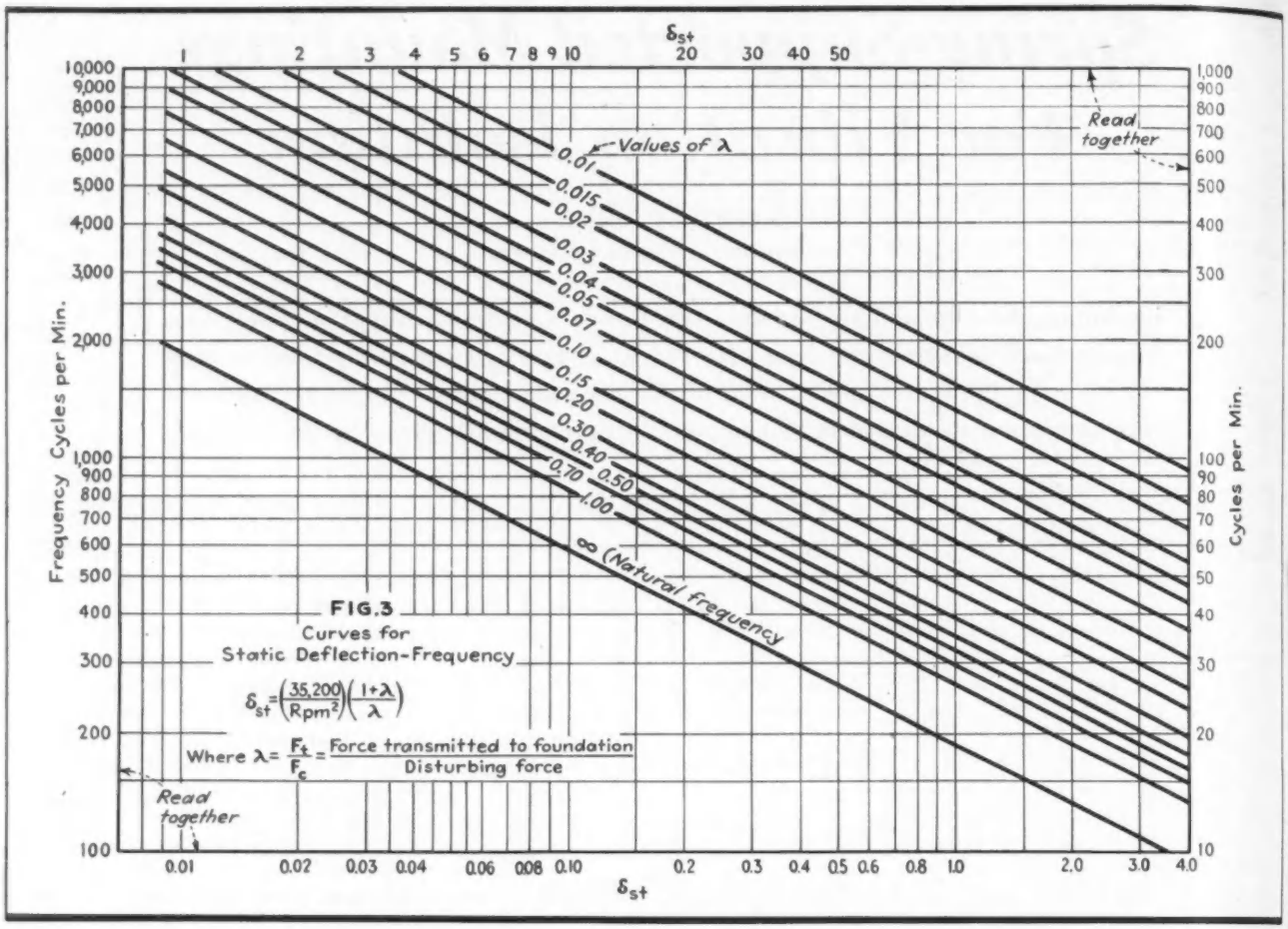


Fig. 3—Curves for static deflection versus frequency for calculating required statical deflection of spring directly from r.p.m. and percentage of isolation.

speed of operation. Let F_c be a centrifugal force resulting from an unbalanced weight W_o located at radius r from the axis of the rotor. Force F_c is then equal to the mass times the central acceleration $\omega^2 r$, that is

$$F_c = \frac{W_o}{g} r\omega^2 \qquad (4)$$

Velocity ω in radians per sec. can be found from the revolutions per minute N by

$$\omega = \frac{N}{60} \times 2\pi \qquad (5)$$

Substituting Equation (5) together with the gravitational constant g equals 386 in. per sec. per sec. into Equation (4) gives

$$F_c = \frac{W_o r\, N^2}{35{,}200} \qquad (6)$$

Substituting for F_c, ω and g in Equation (3) gives

$$x_m = -\frac{W_o r}{\frac{35{,}200\,k}{N^2} - W} \qquad (7)$$

Inspection of Equation (7) indicates that for operation above resonance, or at which the factor

$$\frac{35{,}200\,k}{N^2} < W$$

the amplitude x_m may be reduced by

1. Decreasing the unbalance $W_o r$
2. Decreasing the value of the spring constant k
3. Increasing the rotational speed N
4. Increasing the spring-supported weight W

Designers, of course, will make any changes possible in the vibrating system in order to reduce the amplitude of motion. Often in systems such as spring-mounted motor or engine bases the weight can be increased by means of a heavy cast base plate or concrete slab.

REDUCING THE TRANSMITTED FORCE. The force F_t which is transmitted to the foundation through the spring is found by multiplying x_m by k. That is

$$F_t = kx_m = -\frac{F_c}{1 - \frac{\omega^2 W}{kg}} \qquad (8)$$

Also multiplying both sides of Equation (7) by k gives

$$F_t = kx_m = -\frac{W_o r}{\frac{35{,}200}{N^2} - \frac{W}{k}} \qquad (9)$$

Inspection of Equation (9) indicates that when operation is above resonance, transmitted force F_t can be reduced by

1. Decreasing the unbalance $W_o r$
2. Decreasing the spring constant k
3. Increasing the rotational speed N
4. Increasing the weight W

These are seen to be identical with the conditions for causing a decrease in x_m. Thus whatever is done to decrease the amplitude of the motion will also cause a decrease of the transmitted force.

LIMITING CONDITIONS FOR SPRING MOUNTINGS. From Equations (7) and (9) it is seen that the amplitude x_m and the transmitted force F_t can be made as small as desired by a sufficient increase in W. From Equation (7) it can be seen that no matter how much k may be decreased or the rotation speed N be increased the value of x_m will not approach zero, but an irreducible limit. Thus if k approaches zero and N approaches infinity the amplitude x_m approaches the value.

$$x_m = \frac{W_o r}{W} \qquad (10)$$

Since the velocity of W_o is proportional to its radius r and the maximum velocity of W is proportional to x_m, Equation (10) will be recognized as

the equation from dynamics for equilibrium of momentum.

PERCENTAGE OF ISOLATION. Spring mountings are usually spoken of in terms of "percent of isolation." That is if a mounting transmits but 10 percent of F_c to the foundation it is said to give 90 percent isolation, and so on. It is convenient to use the following ratio λ in solving problems.

$$\lambda = \frac{\text{transmitted force}}{\text{disturbing force}} = \frac{F_t}{F_c} \qquad (11)$$

Where δ_{st} is the statical deflection of the spring caused by W, it is also customary to let

$$\delta_{st} = \frac{W}{K} \qquad (12)$$

Substituting in Equation (3) gives

$$\frac{\omega^2 \delta_{st}}{g} = 1 + \frac{F_c}{F_t} = 1 + \frac{1}{\lambda} \qquad (13)$$

Substitution of previous values for ω and g gives

$$\delta_{st} = \frac{35{,}200}{N^2}\left(1 + \frac{1}{\lambda}\right) \qquad (14)$$

Equation (14) permits the calculation of the required statical deflection δ_{st} of the spring directly from the r.p.m. N and the percentage of isolation. Equation (14) is solved graphically by the curves of Fig. 3 for various value of λ, the transmission factor.

As previously mentioned should the frequency of the disturbing force be the same as the natural frequency for free vibration the operation is said to take place at resonance or at the critical frequency. Here the amplitude as well as the transmitted force theoretically become infinite. Accordingly let λ equal to infinity be substituted in Equation (14). Solving the result for N will give the standard equation for critical frequency N_{cr} in r.p.m.

$$N_{cr} = \frac{187.6}{\sqrt{\delta_{st}}} \qquad (15)$$

The curve is also plotted in Fig. 3 for this equation.

The disturbing force can be written as

$$kx_m = \frac{Wx_m}{\delta_{st}} = \lambda F_c$$

substitution for δ_{st} gives

$$Wx_m = \lambda F_c \delta_{st} = \frac{35{,}200\, F_c\,(1 + \lambda)}{N^2} \qquad (16)$$

If the disturbing force is caused by rotational unbalance, Equation (6) may be substituted giving

$$Wx_m = W_o r\,(1 + \lambda) \qquad (17)$$

This equation is in agreement with Equation (10) if the isolation is 100 percent, that is, λ equals zero.

Equations (16) and (17) show how x_m can be reduced by increasing W or vice versa for any degree of isolation.

Numerical Example

Suppose F_c in Fig. 2 is equal to 1,000 lb. and the r.p.m. is 600. Let 90 percent isolation be required, that is, λ equals 0.1.

From Equation (14) the mounting springs must be such as to give a static deflection of

$$\delta_{st} = \frac{35{,}200}{N^2}\left(1 + \frac{1}{\lambda}\right)$$

$$= \frac{35{,}200}{600^2}\left(1 + \frac{1}{0.1}\right) = 1.076 \text{ in.}$$

This value can be checked by Fig. 3. From Equation (16)

$$Wx_m = \lambda F_c \delta_{st} = 0.1 \times 1{,}000 \times 1.076 = 107.6 \text{ in.}$$

Suppose it would be desirable to hold the amplitude to 0.015 in. The weight W then must be

$$W = \frac{107.6}{x_m} = \frac{107.6}{0.015} = 7{,}170 \text{ lb.}$$

This weight W would of course include engine or motor as well as the base. The required spring constant is found to be

$$k = \frac{W}{\delta_{st}} = \frac{7{,}170}{1.076} = 6{,}667 \text{ lb. per in.}$$

From Equation (15) the critical speed is found to be

$$N_{cr} = \frac{187.6}{\sqrt{\delta_{st}}} = \frac{187.6}{\sqrt{1.076}} = 181 \text{ r.p.m.}$$

This value can also be verified by Fig. 3.

When starting the machine the driving mechanism must be capable of increasing the speed rapidly when passing through the critical range in order to prevent large amplitudes from building up. Some times spring deflection is prevented by clamping W until the speed is well above the critical.

Using the foregoing data let the disturbing force be caused by rotational unbalance, and let the system be suspended by springs attached at the four corners of W. There will thus be no spring resistance to horizontal motion.

By equation (6) the unbalance $W_o r$ is equal to

$$W_o r = \frac{35{,}200\, F_c}{N^2} = \frac{35{,}200 \times 1{,}000}{600^2} = 97.8 \text{ in. lb}$$

By Equation (10) the amplitude x_m is found to be

$$x_m = \frac{W_o r}{W} = \frac{97.8}{7{,}170} = 0.0136 \text{ in. sidewise}$$

Thus the motion for 90 percent isolation is only slightly greater than the motion obtained when the spring constant is reduced to zero.

For engine bases and many similar types of mounts the springs are in compression. It may then be necessary to use horizontally mounted springs in order to give the requisite lateral stability to the system.

Often the horizontal forces are not as great as the vertical. For example, for vertical engines with the usual counterweights the unbalanced horizontal forces are much less than the vertical.

Control of Segregation In Babbitted Bearings

SIGNIFICANT FACTS pertaining to the control of segregation in babbitt are discussed in a paper which was presented before the American Institute of Mining and Metallurgical Engineers at their recent meeting in New York City.

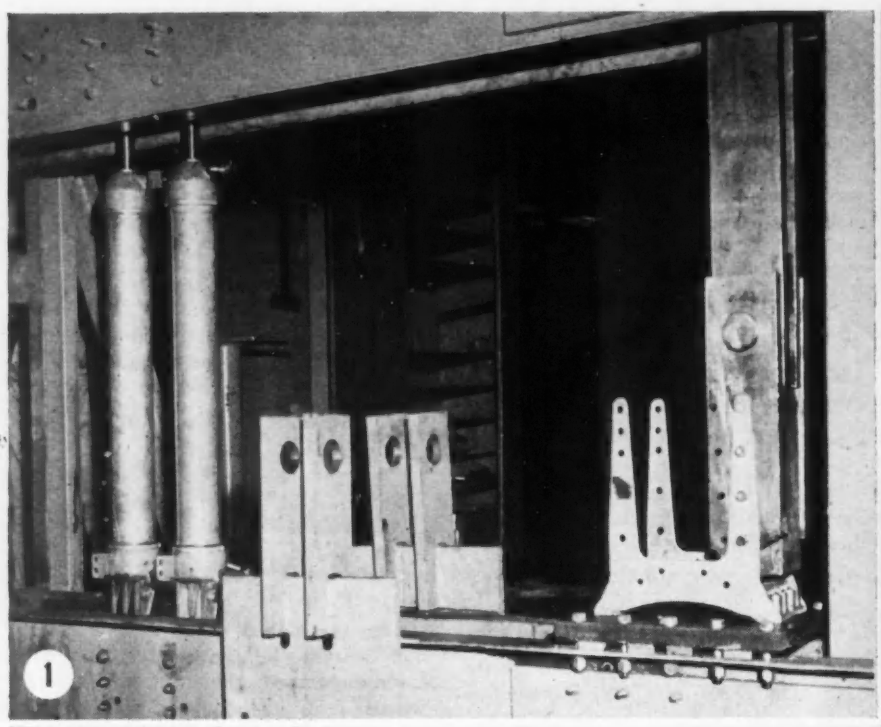
The paper, prepared by T. E. Eagan and W. R. McCrackin, metallurgists of The Cooper-Bessemer Corporation, deals with the effect of temperature and other major factors involved in handling two well known types of babbitt, one a tin-base and the other a lead-base.

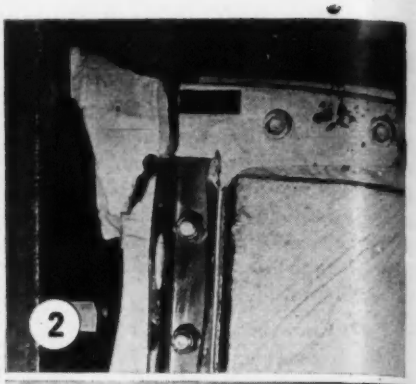
It is emphasized that "to recognize that babbitt is a cast metal will go far in controlling the segregation experienced in making a good serviceable bearing." In the static casting of bearings, careful consideration should be given in designing the metal backing against which the babbitt is cast.

"It has been the practice for many years to design the backing to meet the requirements of stress and fits. Most backs are made of steel, iron or bronze which gives unlimited play to the ingenuity of the designer in making this shell act as a stress member instead of a backing to hold the babbitt. In consequence, dovetails are usually put in to hold the babbitt in place. Subsequent failure is usually attributed to poor babbitt or the wrong composition of babbitt, whereas, the real cause for failure can be traced back to segregation. Without the segregation, the babbitt would amply serve its purpose."

One simple solution that in some instances has brought about a phenomenal improvement in the life and load-carrying ability of a bearing, consisted of casting the babbitt on a separate simple cylinder of uniform cross-section, using it as an insert in the bearing cap.

The practice of cleaning and tinning bearing backs for diesel engines and compressors produced at the Cooper-Bessemer plants was mentioned as being so successful that the dovetail type of design is being supplanted with those of uniform cross section as fast as the drawings can be changed.





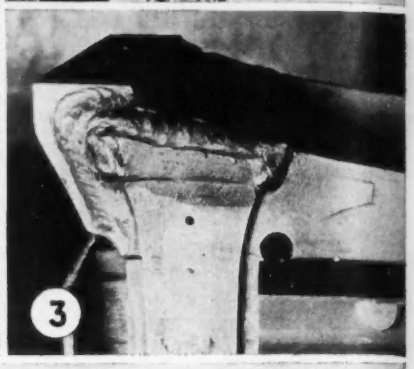


Fig. 1 — Outer wing-root fitting in hydraulic testing jig carried only 131 applications of load in spite of the fact that it was designed by ordinary methods of stress analysis to carry 140 percent design load in static test. Fig. 2 — Outer wing-root fitting after failure from 131 applications of load. Fracture passes through a point of abrupt change of section and a stress raising hole. On underside of plate is a weld bead, which is especially susceptible to fatigue. The fracture has the typical crystalline appearance with no detectable reduction of area. Fig. 3 — Failure of this reinforced outer wing-root fitting occurred after 570 applications of load.

Fatigue Tests of Parts Made Basis for Design

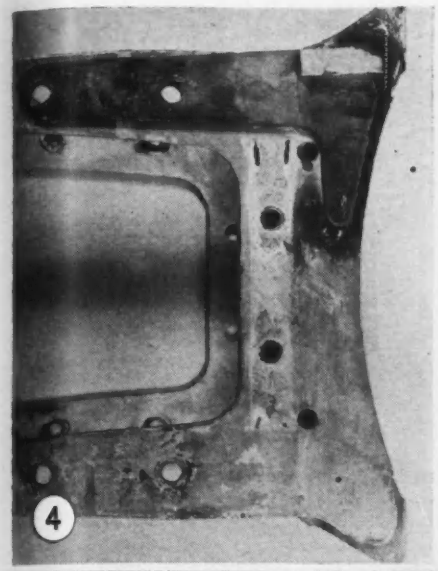
H. O. BOYVEY, *Chief of Development Laboratories*
Consolidated Vultee Aircraft Corporation

Fatigue testing methods were developed to provide design criteria when certain aircraft parts failed, even when designed to carry a static load 50 percent greater than the maximum expected load. "Fatigue limits" based on performance of representative parts, and arbitrary estimates of fatigue loads establish safe values for design.

A LEADING cause of failure in machine and structural members subject to fluctuating loads has been fatigue. Speed-up of design brought about by the war has increased fatigue failures. It has been found that structures capable of carrying more than 150 percent static design load sometimes fail in service because of fatigue stressing. Hence the importance of devising means to determine weaknesses of parts that are to be subjected to fatigue stresses.

The endurance limit stress of most materials used in aircraft construction is low, being in some instances only 25 percent of ultimate strength. Fortunately it is not necessary to use such criteria for design because most structural parts are subjected to an uncertain but not unlimited number of repeated load cycles above the endurance limit stress. The term "fatigue limit" is used to represent the magnitude and number of stresses that can be applied without failure during the normal life of the structure.

The problem is complicated by the fact that two variables are involved: Stress magnitude and number of ap-



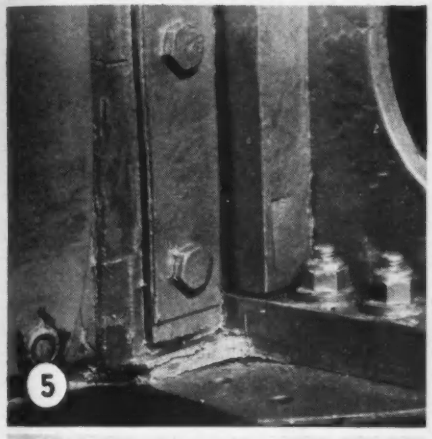
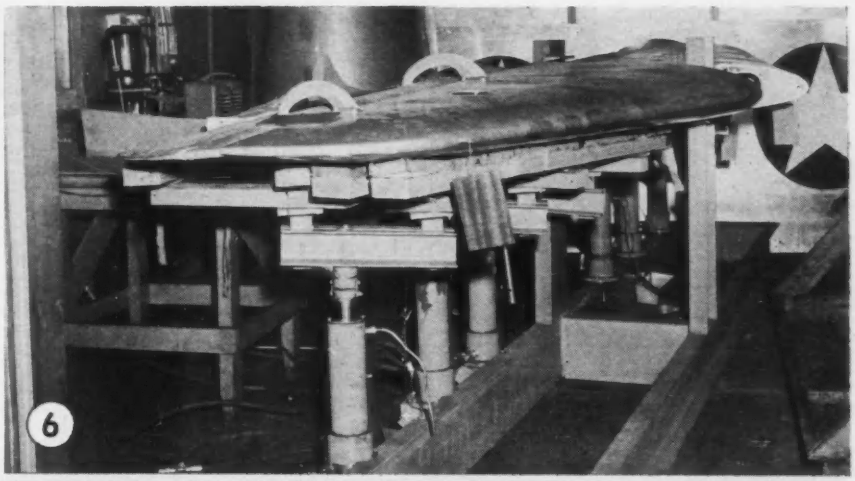


Fig. 4 – An L-shaped reinforcement gave the outer wing-root fitting sufficient fatigue strength. This fitting did not fail until maximum load was applied 258 times more than the required 2,000 times.

Fig. 5 – A horizontal stabilizer is a typical plywood part that successfully passed static test requirements, carrying 170 percent design load, but failed to stand up in fatigue and weathering tests. Failure occurred after load was applied 245 times. The test followed 5 weathering cycles without load.



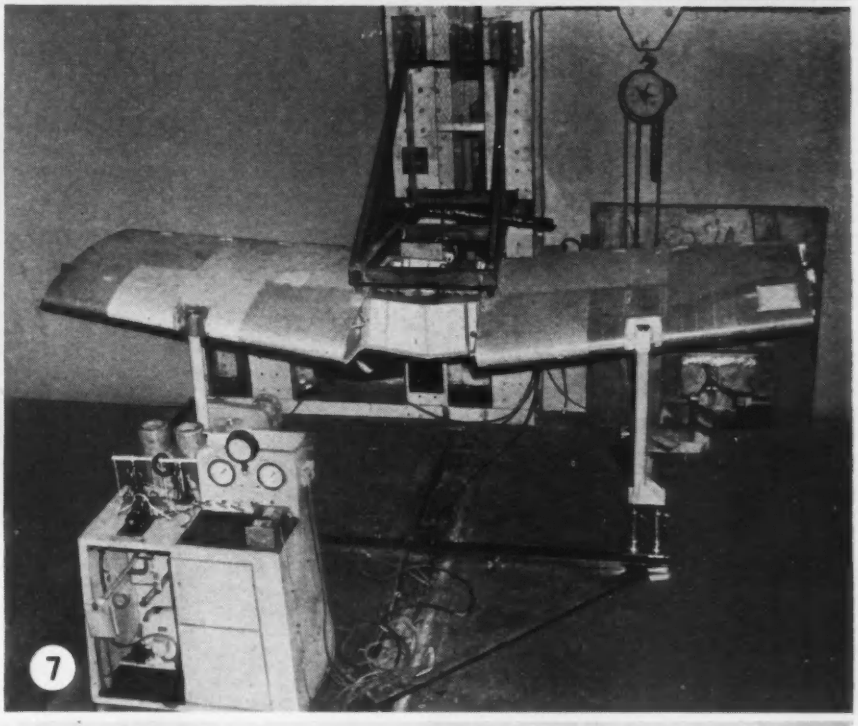


Fig. 6 – Redesigned plywood stabilizer, after being subjected to 5 weathering cycles while repeated applications of load were imposed, carried 33,000 load applications without failure. Fig. 7 – A typical automatically operated fatigue test set-up using double-acting hydraulic jacks with 7.1 sq. in. piston area fed through flexible tubing from a 3,000-lb. "load maintainer" consisting mainly of a variable delivery, compensator pressure control pump, a 5-hp. electric motor, two Beggs type valves, which allow two separate lines of pressure to be used at one time, cam-driven control valves, automatic shut-off valves, and gages. Quick-change gearing in the valve control mechanism permits changing speed of loading cycle. Most fatigue tests are run at 1 to 5 cycles per minute.

plications of "above endurance limit" stresses. A known relationship exists between the allowable number of stress reversals and the magnitude of stress applied and is often represented by the well-known S-N curve. The Moore rotating-beam fatigue machine test is satisfactory for obtaining basic comparisons between materials, which can be plotted in S-N curves, but does not predict the behavior of a fabricated part because of the influence of forging practice, handling, workmanship and other factors on fatigue resistance. "Stress raisers," such as abrupt changes of section, cut-outs and rivet holes, cannot always be avoided.

Service data are the most valuable source of information for establishing criteria that insure trouble-free parts. Representative parts, which are known to have a satisfactory service life, are fatigue tested to failure by repeated applications of the maximum load expected to be imposed on the structure in service. This load is two-thirds of the design load. The structure must be capable of carrying the design load but need not do so without permanent set or damage to the structure. The number of cycles without failure is then an index of the upper limit of the required resistance to fatigue. Since this figure may be larger than necessary, a rational standard of comparison is reached by testing other representa-

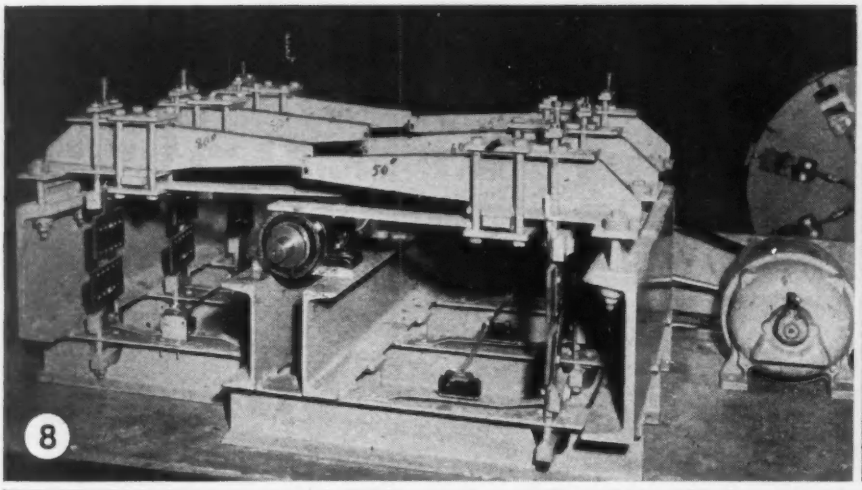


Fig. 8 – This mechanical fatigue testing machine handles 6 specimens and applies 530 loads of 0 to 600 lb. per specimen per minute in one direction only. Load is applied by a leaf spring resting on an eccentric. An electric time clock is stopped at each failure. This machine is generally used to obtain shear fatigue data on spot welds, rivets and other attachment methods.

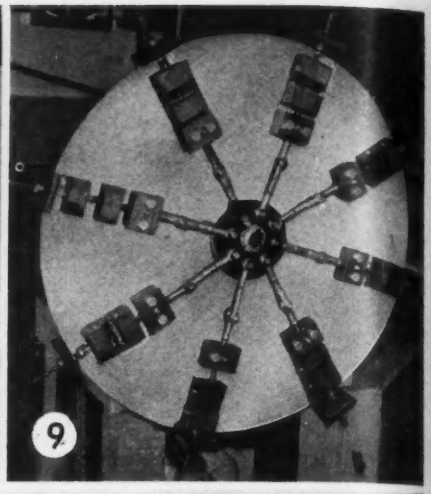
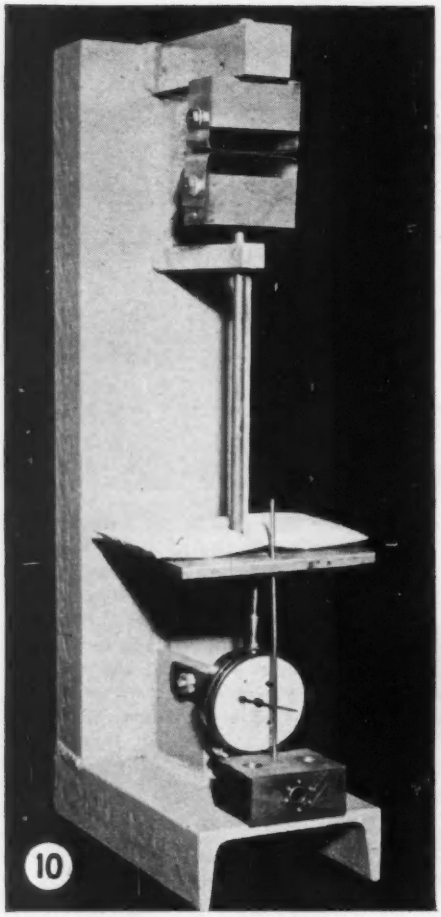


Fig. 9 – Rotary eccentric tension fatigue machine developed by Convair engineers for studying tensile fatigue characteristics of spot welds, rivets and other attachment methods. Loads are controlled by adjustment of the screw supporting one end of the specimen.



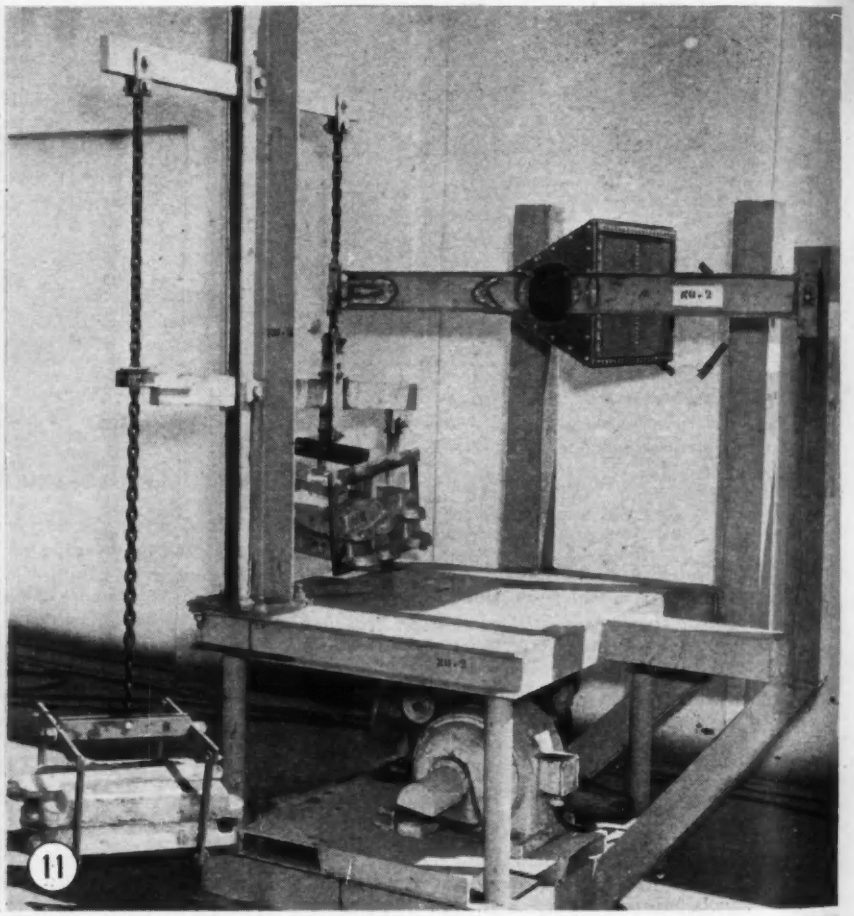
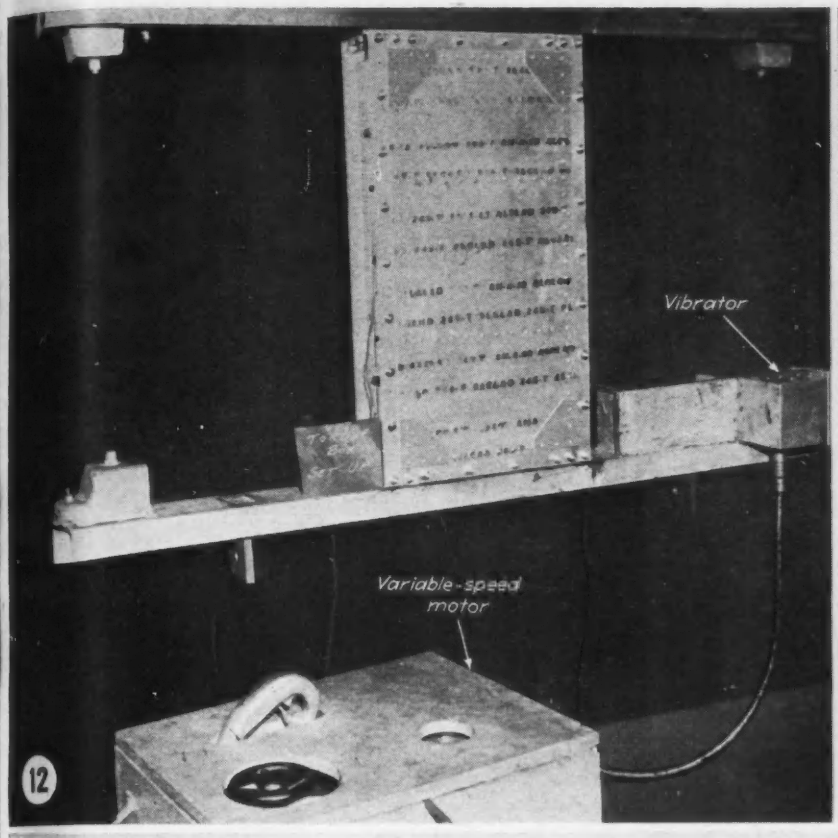


Fig. 10 – Jig for calibrating test specimens from the rotary tension fatigue machine for load and deflection. Fig. 11 – Fatigue test machine developed to study basic sheet materials. Test specimens are mounted in the 3-cornered frame, which is connected to the actuating beam on one end and to the main frame of the machine on the other. The stationary end of the box is mounted on a shaft that allows the box to rotate in the bending direction but not in the torsional direction, thus subjecting specimens only to pure shear. The three-cornered box subjects each panel to the same shear load even though the relative rigidity of the panels may be different. Thus different materials of different gages can be tested at the same time. The machine is relatively slow, reversing the load at 50 cycles per minute.



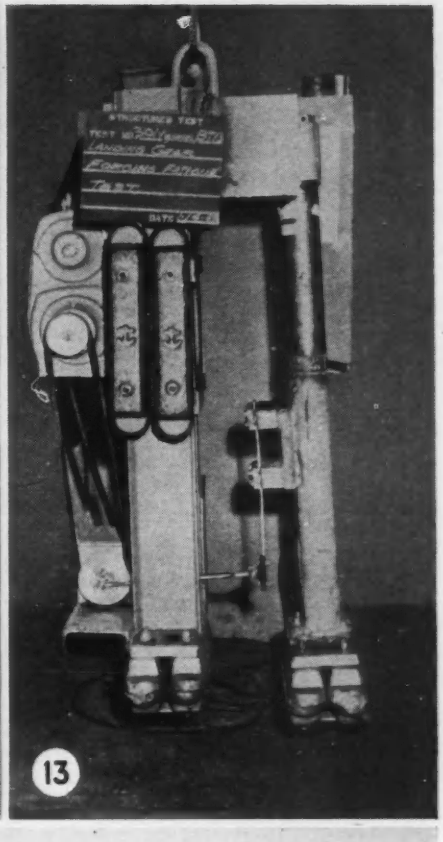


Fig. 12 — A high-speed method for fatigue testing two panels at once. Weights are attached to beams at each end of the panels. Load is applied by a variable-speed vibrator driven through a flexible shaft. A Mall Universal motor and Variac transformer are used to control speed, which is usually about 2,000 cycles per minute. The vibrator head consists of a pair of Micarta gears with eccentric weights arranged to confine vibration in one plane. Stress and frequency are recorded by means of electrical strain gages and specially built dynamic equipment. The machine is suspended by a cable. Stress and frequency recording equipment used with test machine consists of a standard Brush amplifier and recorder, a pre-amplifier and mixing panel and a calibration unit. Fig. 13 — High-speed fatigue testing a landing gear forging. The forging is mounted in the upper yoke and has a dummy landing gear strut installed with a suitable mass at the lower end and a leaf spring connection to the exciting mechanism. The entire machine is suspended by a cable, thereby eliminating noise and preventing the building from shaking. Load is controlled by weight and frequency of masses involved. Frequency of 750 cycles per minute while imposing a bending moment of 94,500 in. lb. in each direction has been attained. Required power input is small if run at resonant frequency. This can be done by proper selection of weights and rigidity of the system when the apparatus is set up for test.

tive parts that have failed in service.

The relatively small amount of factual data available often makes it necessary to establish standards arbitrarily, based upon available facts. For example, an outer wing part of a trainer-type airplane encounters maximum loads in flight, but not in ordinary flight under normal weather conditions. Inexperienced pilots, however, may impose severe flight loads on the wings. If service life of this airplane is assumed to be six years, flying 300 days per year, service life will be 1,800 flying days. Assuming that maximum load is reached on an average of once per flying day, total applications will be 1,800. Since the reasoning is arbitrary the criteria is boosted from 1,800 to 2,000 for design purposes.

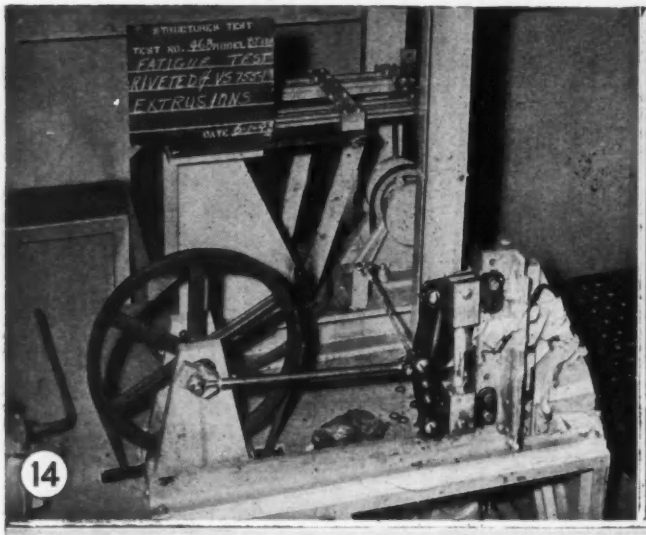
If the part in question had been a component of the landing gear, a different analysis would be made because trainer aircraft alighting gears are subjected to severe service. Expected "mortality rate" is considered higher than for wing parts. Assuming three severe landings per flying day, there would be 5,400 maximum load applications, which are arbitrarily increased to 6,000 for design purposes.

If the airplane under consideration were a commercial transport, landing gear criteria would be reduced and wing considerations would be increased, because airline pilots are more skillful in landing and because of the necessity of flying in all kinds of weather.

Experience of Consolidated Vultee in testing an outer panel wing root fitting for its training cantilever monoplane illustrates how this arbitrary method of arriving at a fatigue limit affects design. Figs. 1 to 4 show methods and results of fatigue tests and redesign, which changed the number of applications of load before failure from 131 to 2,258, a figure that is considered satisfactory.

Increased use of plywood in aircraft has brought out interesting fatigue data. Plywood structures like those of metal often carry high static loads but fail from fatigue in service. A standard method of testing plywood aircraft components has been devised by Consolidated Vultee engineers. This procedure takes into account changing weather conditions as well as fatigue from repeated applications of load. In these tests, plywood parts are subjected to five cycles of weathering while the repeated loads are being applied. Each cycle consists of:

1. A "wetting period" of 48 hr. during which air temperature is 90 deg.



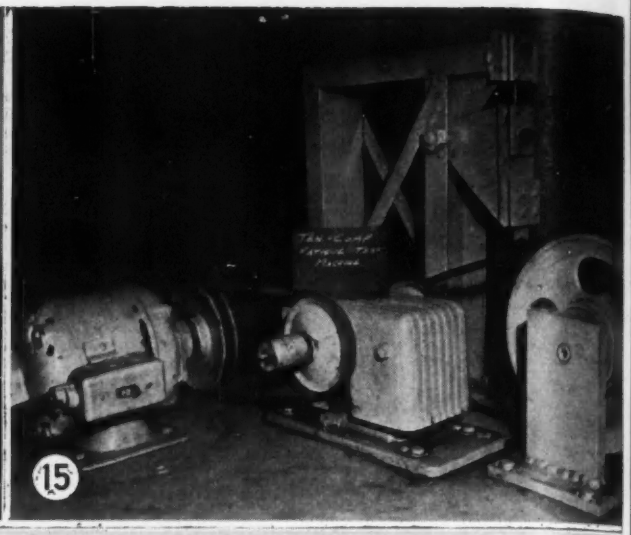
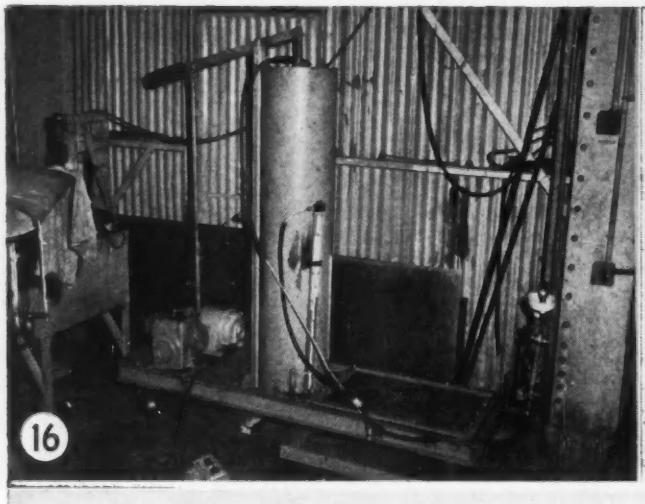


Fig. 14 — Toggle-type machine in foreground is used for fatigue and wear tests of bushings, bearings, joints, and other parts. The machine in the background is a simple eccentric type for fatigue testing materials in bending. It is often used for testing various plywood paint finishes under flexure. Fig. 15 — Notch sensitivity and basic fatigue data under conditions of complete reversal of stress are obtained with this machine.



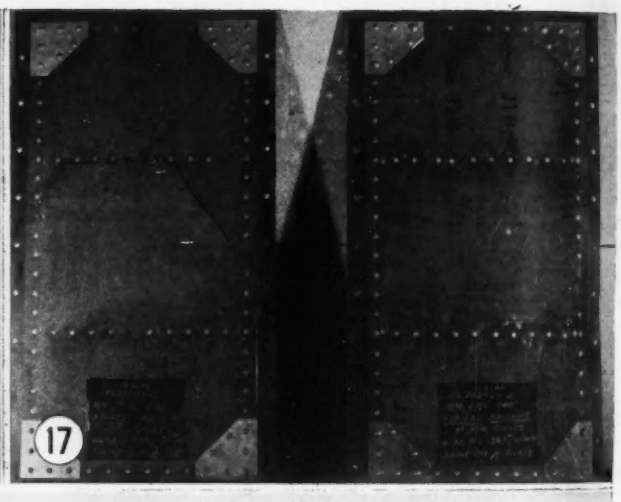


Fig. 16 — Light-weight non-structural or semi-structural surfaces, such as the aft portion of a wing, are fatigue tested by this machine consisting of a piston type air pump operating at 60 strokes per minute to apply a pressure of 288 lb. per sq. ft. to the test panel.

Fig. 17 — Typical failures of magnesium sheet panels from fatigue tests. Effects of changing material and gage, riveting or welding methods, backing or stringer design, and other variables can be determined rapidly by this method.

F. and relative humidity is 100 percent.

2. A "cold period" in which temperature is held at —25 deg. F. for about 24 hours.

3. A 24-hr. "drying out period" during which temperature is held at 160 deg. F. and relative humidity at 10 percent.

Figs. 5 and 6 illustrate the application of fatigue tests to a plywood horizontal stabilizer and show results of tests that enabled redesign with assurance of withstanding service load conditions. It is realized that the exterior finish of plywood parts is subject to some weathering deterioration and that warping and cracking are likely to occur under certain conditions of temperature and humidity.

The equipment used in performing these typical fatigue tests of outer wing-root fitting and plywood stabilizer is of special design, the loads being applied by means of hydraulic jacks and automatic controls such as illustrated in Fig. 7. Since the speed of the loading cycle of hydraulic equipment is relatively slow, it is used usually only when loads are too large to be handled by mechanical equipment. It can be used for static tests also by cutting out the motor-driven valve and using a hand-operated valve.

When loads are small and a large number of specimens are to be tested, mechanical machines are used. Figs. 8 to 17 illustrate mechanical fatigue testing machines and accessory equipment. The ordinary table top vibration machine, which is too well known to need description, and is not illustrated, is often used for fatigue testing.

All of the fatigue test described here give immediate comparative results that can be applied directly in design or redesign. They permit quick checks that minimize the time for design.

Electric Braking Methods— Reversal of Power by Plugging

E. H. HORNBARGER
Control Engineer, Westinghouse Electric & Manufacturing Company

Basic considerations involved in the selection and application of plugging arrangements to conventional d.c. and three-phase a.c. motors to stop quickly or retard the motor and its driven load. Chart summarizes the advantages and disadvantages of the plugging method with diagrams of different schemes for control.

ACCURATE retarding and stopping of motors is important in many industrial drives. Elevators must slow down to a smooth stop for safety and comfort. Crane loads must be kept under control to protect the crane, the material being handled, and the operating personnel. Metal-working planers must be stopped quickly and accurately not only to protect tools and work, but also to meet the requirements of modern high-speed production. Time spent in stopping and reversing is time lost. A productive tool cannot afford to coast to a stop.

Some special applications rely on accurate stopping to meet proper functional interlocking with which split-second timing is essential. Continuous processes may be interrupted if one element in the process is improperly timed. In emergencies, some machines used in rubber mills, must be stopped instantly to prevent injury to operating personnel.

There are two general classes of electric motor braking. The first is the well-known solenoid-operated friction brake of the shoe or disk type. The second is all-electric, the operation depending upon forces inherent in the motor itself. In industry, all-electric braking is often used in combination with solenoid-operated or mechanically set brakes. Applications which require a holding force when shut down must be provided with some type of mechanically-operated brake, because all-electric braking is not effective when the motor is at rest. The holding force must be independent of the power supply, the state of rest or motion of the motor, and its load.

There are three methods of all-electric braking:

PLUGGING is the application of power in the reverse direction which causes the motor to exert torque in a direction opposite to the direction of rotation and thus quickly bring itself and its driven load to a stop. When the motor reaches zero speed it can be arranged to stop or to reverse and accelerate in the opposite direction.

DYNAMIC BRAKING removes the motor from the power supply and connects the motor as a generator, loading itself on a bank of resistors or on part of its own winding.

REGENERATIVE BRAKING is similar to dynamic braking in that the motor acts as a generator, but in regenerative braking the motor is not disconnected from the line. It is loaded by feeding its generated power back into the line.

Braking torque, stopping time, resistor design and excitation requirements can be calculated only when all facts concerning the motor and its driven load are known. But there are some basic considerations that may be helpful in applying electric braking to conventional d.c. and three-phase a.c. motors.

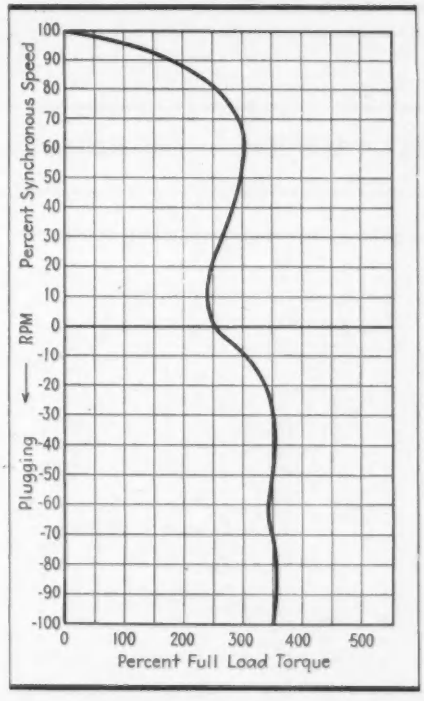


Fig. 1—Speed-torque curve of a typical 7½ hp. squirrel-cage induction motor showing the torque developed when the motor is plugged on full voltage.

Plugging Methods

The main advantages of plugging are cheapness and simplicity, also it is the quickest way to stop a motor and its load. The chief disadvantage is its absolute dependence upon the power supply. When power fails, there can be no plugging. The time required to plug a motor to "stop" depends upon the inertia of the rotating mass and the plugging torque of the motor.

A.C. SQUIRREL-CAGE INDUCTION MOTORS. Plugging is the simplest, quickest and most widely used method of securing all-electric retardation of squirrel-cage motors. No d.c. source and little extra control are required. But the control unit must include a reversing (plugging) contactor. Plugging a motor and its driven load imposes considerable strain on the motor windings, therefore, they must be able to withstand such duty. Fig. 1 shows torque developed in a typical 7½ hp. squirrel-cage motor when plugged on full voltage. The capacity of the power system must also be capable of absorbing the peak current required for plugging.

A cushioning effect can be provided by the use of series resistance in the plugging circuit, to reduce the shock to machine and power system. But this arrangement reduces the plugging torque and increases the time required to stop and to reverse the motor.

When it is imperative that the motor come to a stop without reversal, a simple device known as a zero-speed switch can be linked to the motor shaft,

or other rotating part of the machine. The zero-speed switch is a centrifugal switch the contacts of which, though open at rest, close as the motor begins to rotate. The control circuit is so arranged that when the motor is operating and the stop pushbutton is depressed, the line contactor drops out and through a back contact sets up a circuit in series with the now closed contact in the zero-speed switch to energize the closing coil of the reversing contactor.

When the reversing contactor closes, the motor attempts to reverse its direction of rotation and rapidly decelerates to a stop. As the motor nears a state of rest, the zero-speed switch opens its contacts, deenergizing and dropping out the reverse contactor, thereby preventing reversal. The control can be arranged as desired to provide "plug stop," or normal stop in either direction of rotation. Also, the control can be arranged to provide a continuous "start-plug-start-plug" operation so long as an "inching" push-button is held down, resulting in a ratcheting-type of creeping speed.

Where the use of a zero-speed switch is impracticable, similar results can be obtained by the use of a relay to govern the plugging time. Such a relay must be accurate and consistent in its time of operation, and the inertia of the parts that are to be stopped must remain constant, otherwise the results will be erratic and unsatisfactory.

A.C. Wound - Rotor Induction Motors. Plugging has long been used for wound-rotor induction motor drives. When the master switch is manually operated the operator throws it in the opposite direction, reversing the power supply to the motor. The plugging torque at any speed can be increased or decreased by adjusting the value of resistance in the motor secondary circuit.

If there is considerable resistance in the motor secondary circuit, and if the load is over-hauling, the counter torque may be sufficient to retard but not sufficient to stop the motor and its load. In the design of hoists and cranes, this condition is known as counter torque lowering. If reversing is to be prevented the operator must move the master switch to the off position when the motor comes to rest after plugging.

Since plugging of wound-rotor motors is used principally on reversing applications, no special or extra control equipment is necessary. In plugging from full speed, the motor secondary voltage is double its normal locked rotor value. Resistors, control devices, and motor windings must therefore be insulated for this higher voltage. Also, the motor windings may require special bracing.

Synchronous Motors. Although synchronous motors are used only on constant speed applications and are not intended for frequent starting and stopping, they act essentially as squirrel-cage motors during their starting period and can be plugged to a stop. The d.c. field is deenergized during the plugging period. Plugging of synchronous motors is rare, since for quick stopping they lend themselves more readily to the smoother and gentler retardation secured through dynamic braking. The plugging feature of the control is the same as for a squirrel-cage motor, that is, a zero-speed switch is used for a quick stop without reversal. The motor design should be carefully checked as to its ability to stand such service.

D.C. Motors. Since d.c. motor control is easily arranged for dynamic braking, plugging alone is rarely used except with series-wound mill motors. When d.c. motors are plugged, additional resistance must be used to prevent excessive current which might injure the equipment. D.c. controllers for reversing service usually provide dynamic braking as a standard feature and are often provided with anti-plugging relays to prevent reversal until the motor decelerates to a speed where reversal is safe. Metal-working planer controllers use a combination of dynamic braking and plugging to secure quick reversal. The motor is first slowed down by dynamic braking before its connections are reversed.

Where plugging alone is used to secure quick stopping of d.c. motors, a zero-speed switch is used, like that for a.c. motors, but extra resistance must be included in the plugging circuit.

The accompanying chart summarizes the advantages and disadvantages of the plugging method for retarding and stopping motors. The diagrams include the basic schemes of control and the main connections for each. Overload and other protective relays are not shown in the diagrams.

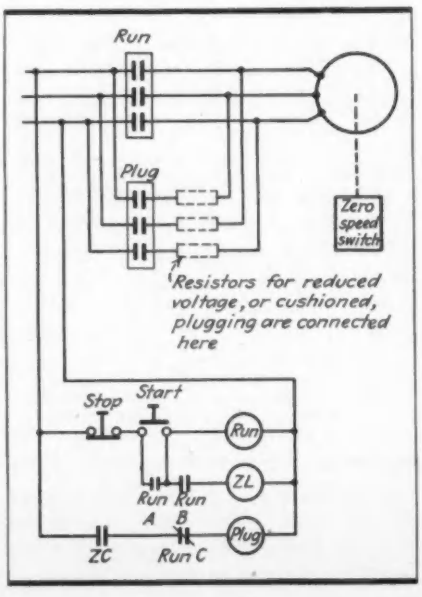
PLUGGING METHOD OF RETARDATION

Advantages

Simple arrangement and installation.
Fast—operating performance.
Free to rotate after stopping in contrast to mechanically set brake.
No friction parts and freedom from maintenance.
Suitable for large motors and severe duty.

Disadvantages

Requires reversing controller.
Requires zero-speed switch or consistent timing relay if reversing is to be prevented.
Power must be available. Cannot plug if power fails.
Power system must be capable of supplying the peak plugging current.
Motor windings may require special bracing.
If motor and its load must positively be held stationary when the motor is deenergized, then additional spring-set or other type of mechanical brake is necessary.



THREE-PHASE SQUIRREL-CAGE MOTOR PLUG STOP WITH ZERO-SPEED SWITCH

Zero-speed switch, linked to and rotating with the motor, is equipped with a lock-out device, the coil *ZL* of which must be energized before the switch contacts *ZC* can close.

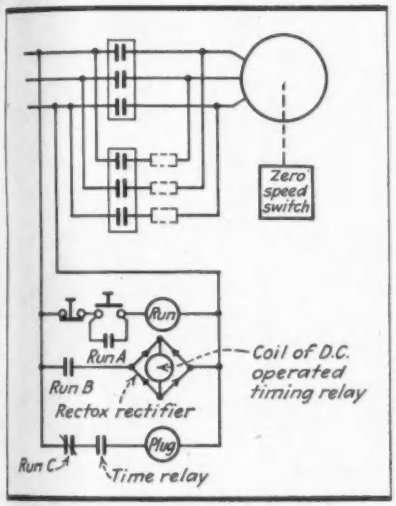
When the start button is depressed, the run contactor closes, and connects the motor to the line. At the same time, auxiliary run contact *A* sets up a holding circuit around the start button and run contact *B* energizes the coil *ZL*. As soon as the motor begins to rotate *ZC* closes, this would energize the coil of the plug contactor had not auxiliary run contact previously opened to prevent such a circuit.

When the stop button is depressed, the run contactor drops out and dis-

connects the motor from the line. Auxiliary run contact *C* closes and energizes plug contactor which closes and connects the motor to the line in the reverse phase direction. As the motor reaches, or nearly reaches, a standstill the centrifugally operated switch *ZC* opens and drops out plug contactor, thereby preventing reversal.

SQUIRREL-CAGE INDUCTION MOTOR PLUG STOP WITH TIMING RELAY

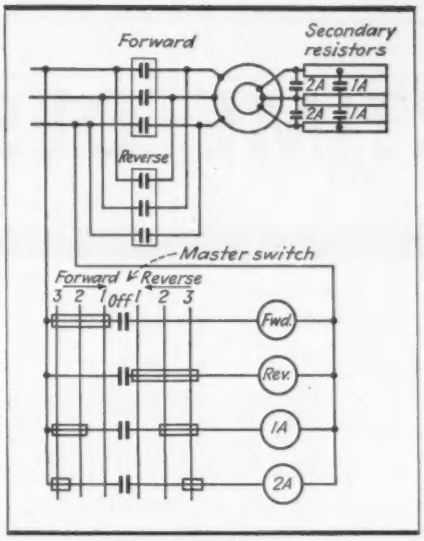
TIMING RELAY is substituted for zero-speed switch. Relay shown is the flux-decay type which picks up instantly but which, when deenergized, drops out after an adjustable time interval. Relay picks up when the run contactor closes and remains closed while the motor is running. When the stop button is depressed, run contactor drops out and the auxiliary run contact *C*



completes the circuit to the coil of plug contactor. Plug contactor closes and reverses the power to the motor. Plug contactor remains closed until the timing relay, which was deenergized by run contact *B*, opens its circuit.

WOUND-ROTOR INDUCTION MOTOR

DIAGRAM shows a 3-point wound-rotor induction motor controller with a manually-operated reversing master switch. In the first point, forward or reverse, the respective contactor closes and starts the motor in the corresponding direction. In the second and third points, accelerating contactors 1*A* and 2*A* close, cutting out the starting resistors in successive steps. To plug, the operator moves master switch handle through its off direction and into one of the opposite operating points. Motor will then be plugged, either with or without resistance in



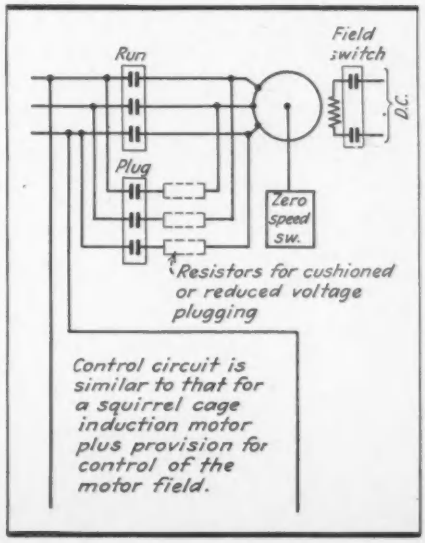
its rotor circuit depending upon the point at which master switch handle comes to rest. If reversal is to be prevented, the operator must move the master switch to its off position as the motor comes to rest.

Controller can be fully automatic, pushbutton operated, making use of the zero-speed switch to prevent reversal, or it can be arranged for reversal. Wound-rotor motors for plugging service are usually required to operate in either direction.

SYNCHRONOUS MOTOR

SYNCHRONOUS MOTOR can be plugged to a stop in the same manner as that shown for a squirrel-cage induction motor. Zero-speed switch, or timing relay, is used to open the plugging contactor as motor comes to rest and before reversal can take place. Control of the motor field is one function of the controller, but it has been omitted from the diagram, except that the field switch contacts are shown.

Resistors are indicated in the plug-


Run
Field switch
D.C.
Plug
Zero speed sw.
Resistors for cushioned or reduced voltage plugging
Control circuit is similar to that for a squirrel cage induction motor plus provision for control of the motor field.

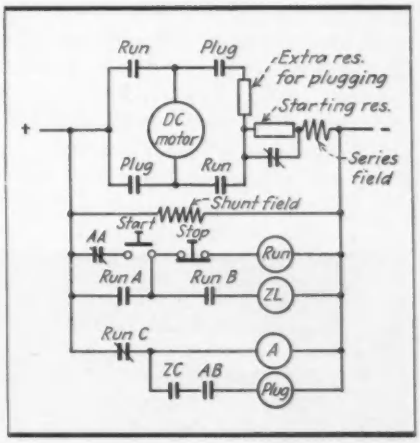
ging circuit, but where the motor is started from reduced voltage taps of an autotransformer, or compensator, it would naturally be plugged on the same source of reduced voltage.

DIRECT CURRENT MOTOR

SIMPLIFIED DIAGRAM shows how a d.c. motor can be plugged to a stop, without reversal, by use of the zero-speed switch. Accelerating timetactor coil *A* is shown energized when the motor is at rest. Its main contacts are open when its operating magnet is energized and therefore the starting resistor is in series with the main circuit to motor. Also the two auxiliary contacts on *A* are closed, *AA* is in series with the start button and *AB* is in series with the coil of the plug contactor.

Depressing start button energizes the coil of the run contactor which closes and starts the motor on reduced voltage. Auxiliary contacts on run perform the following functions. Run *A* sets up a holding circuit around the start button. Run *B* energizes coil *ZL* which will allow the zero-speed switch to close its contacts *ZC* when it begins to rotate. Run *C* opens the circuit to coils *A* and plug. Accelerating timetactor *A* drops out after a definite time and closes its main contacts, thus shorting out the starting resistance.

When the stop button is operated, run drops open closing its normally closed auxiliary run contact *C* which



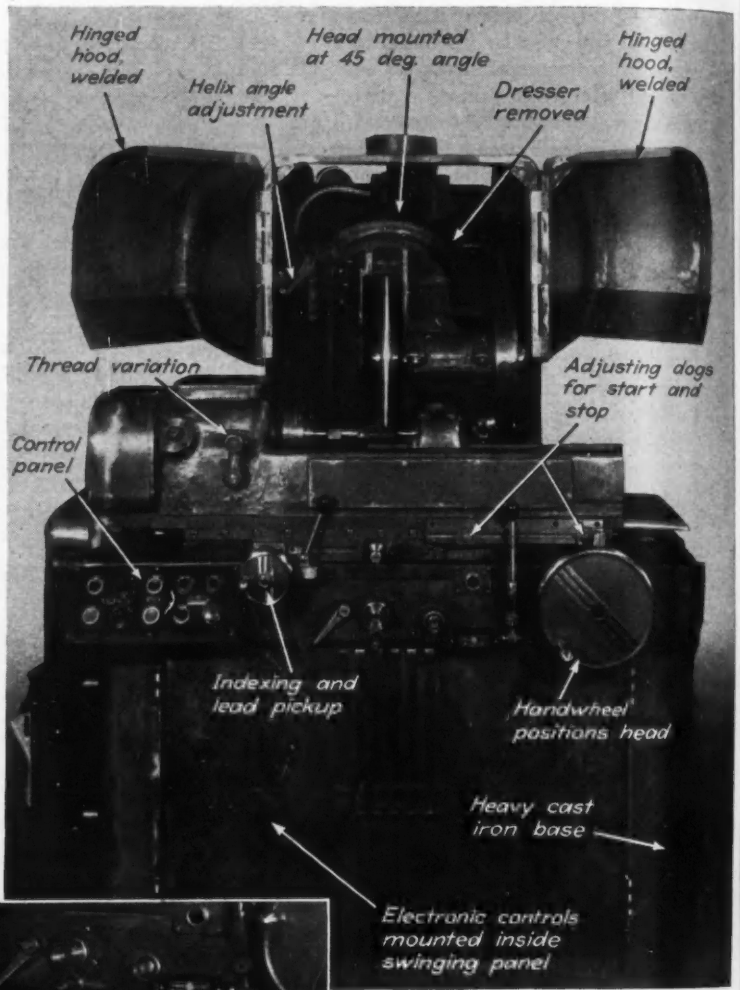
again picks up *A* inserting the starting resistance into the motor circuit. Auxiliary contact *AB* closes. The plug contactor, the coil of which is now energized through run *C*, *ZC* and *AB*, closes and plugs the motor. Note that an extra resistance is used to limit the plugging current and that in plugging the starting resistance remains in the circuit.

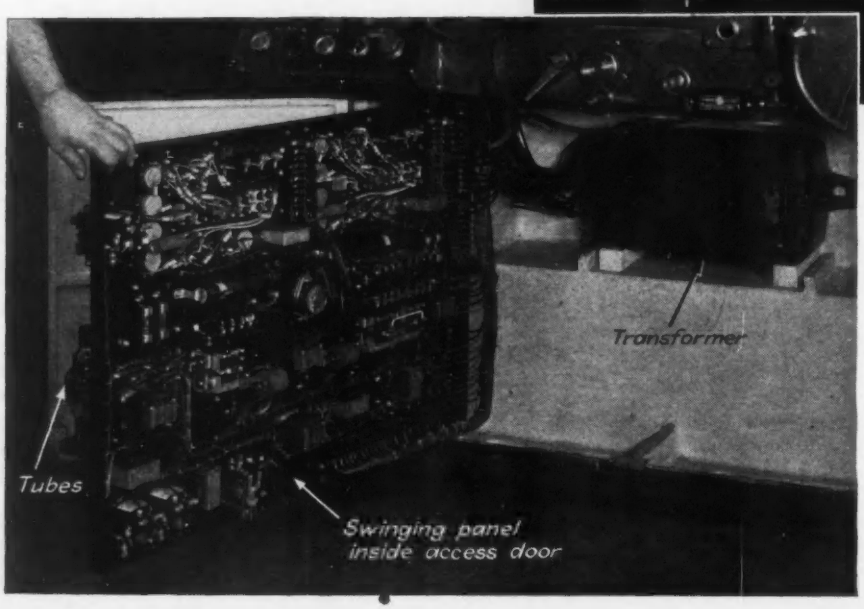
As the motor nears standstill, *AC* opens and deenergizes the coil plug, removing power from the motor.

WARTIME DESIGNS

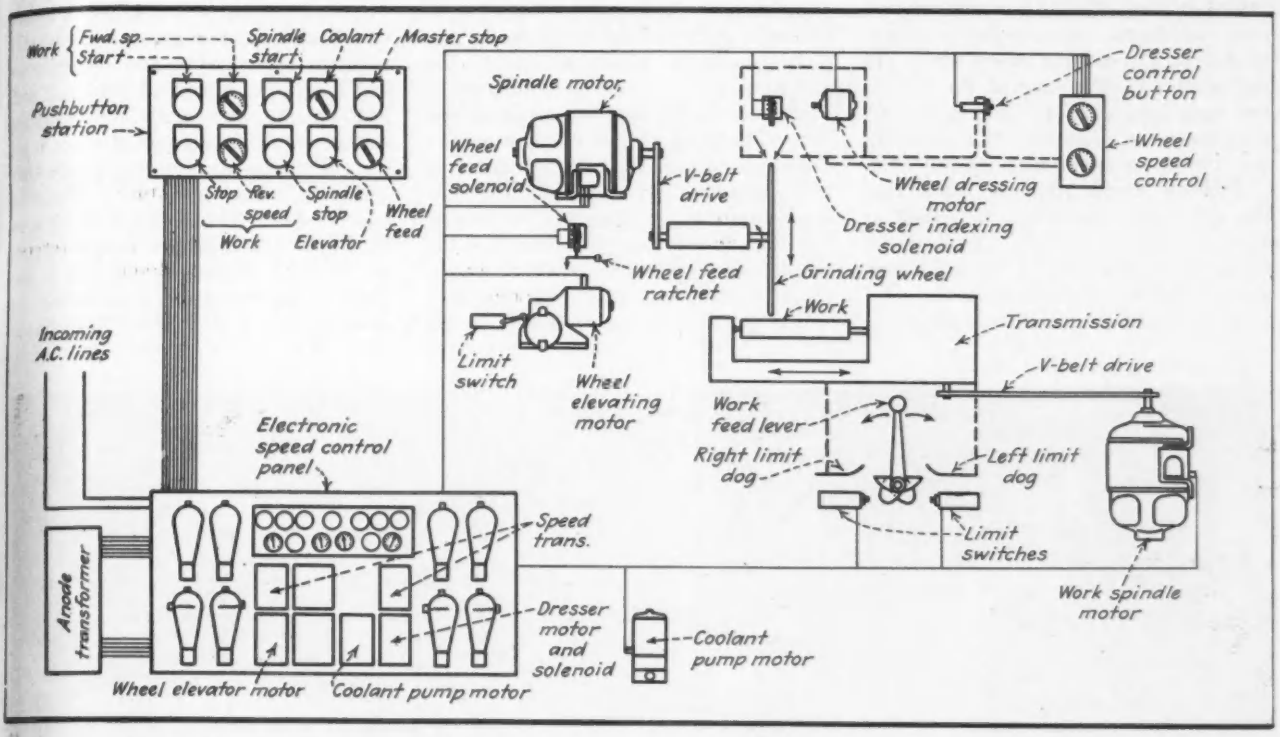
Electronic Drive Used in New Thread Grinder

INFINITE, stepless variation of grinding speed—both forward and reverse—and constant peripheral speed regardless of wheel wear are obtained electronically in the new Dalzen thread grinder by the use of the Thy-mo-trol drive developed by General Electric. Other features of the new machine are full vision at the point of work, easily accessible operating controls, and automatic dressing for any thread form. The machine is mounted on a heavy cast-iron base which absorbs vibration. The base is compartmented to receive accessories. Coolant tank design has baffles to provide slow precipitation of grit and foreign matter.

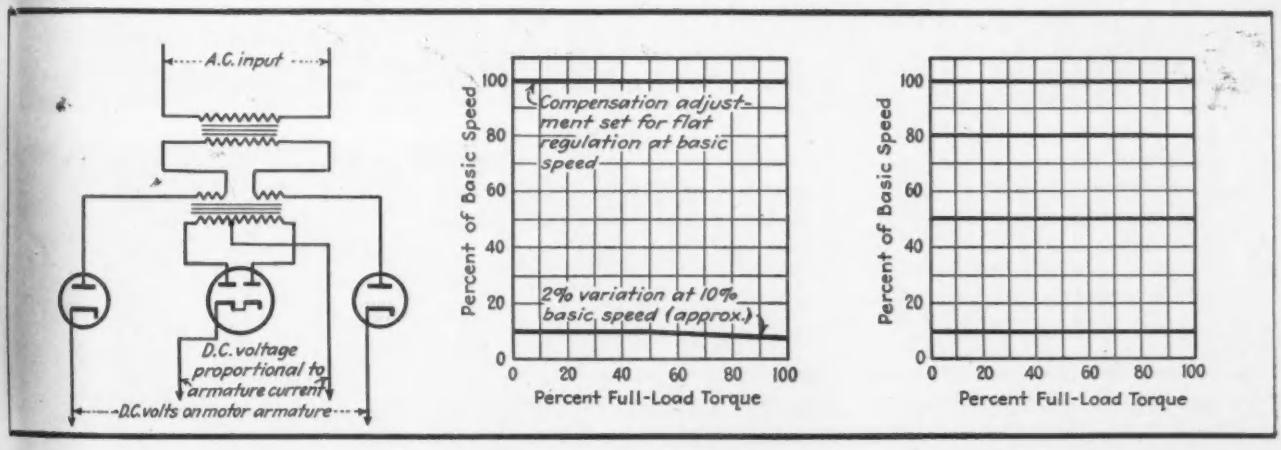




Electronic control is mounted on a swinging panel inside the front access door. This facilitates assembly and simplifies servicing. Electronic tubes and units are mounted on front side of hinged panel; all wiring is mounted on rear of panel. Transformer is housed in a recess provided in the base casting.



Schematic circuit showing electronic connections to various motors and mechanical interlocks controlling motor operation. Grinder spindle and work spindle motor are 1 hp. d.c. Grinder spindle speed range is 1,200 to 3,500 r.p.m.; work spindle speed range is 135 to 3,450 r.p.m. Coolant pump motor ¼ hp., wheel elevating motor ⅙ hp. and the fractional hp. dresser motor are also connected through the electronic panel, as indicated above.


A.C. input
D.C. voltage proportional to armature current
D.C. volts on motor armature
Percent of Basic Speed
Compensation adjustment set for flat regulation at basic speed
2% variation at 10% basic speed (approx.)
Percent Full-Load Torque
Percent of Basic Speed
Percent Full-Load Torque

Two electronic rectifiers are incorporated in the panel to supply voltage to motor field and armatures. Separate control of armature and field voltage gives complete control over d.c. motor performance. When power is applied to the anode transformer, the tube filaments begin to heat. A timer on the electronic panel governs delay before the thyratrons supply current. This time interval helps insure long tube life. When the timing cycle has elapsed, field voltage is automatically applied to the motor. As the start button is pressed, a line contactor is closed and power from the armature rectifier is applied to the armature. After the line contactor is closed, the control of the motor is entirely electronic.

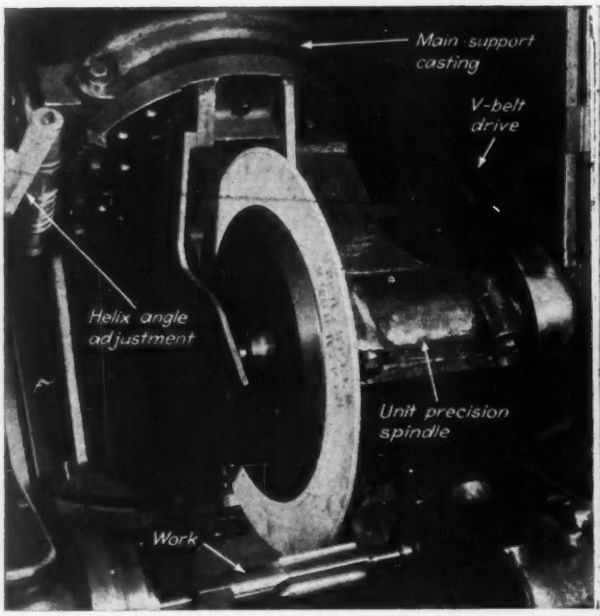
Because there is no back-voltage when the motor is at rest, armature current will be high if full voltage is applied. This current must be limited to prevent excessive current and torque peaks which might injure the load on the motor. The electronic device measures the armature current, and automatically limits the d.c. output of the controlled rectifiers to a safe value through the use of a current transformer with primary windings connected in series with the tubes, which supply armature current. Transformer is so designed that the pulsating current passing through its two primary windings causes an a.c. voltage to be generated in its secondary. This a.c. voltage is rectified, and the d.c. voltage obtained is proportional to the armature current which flows through the rectifier. This voltage is connected into the electronic control circuit in such a way that it limits the voltage output of the thyratrons to that required to maintain the armature current at the preset value.

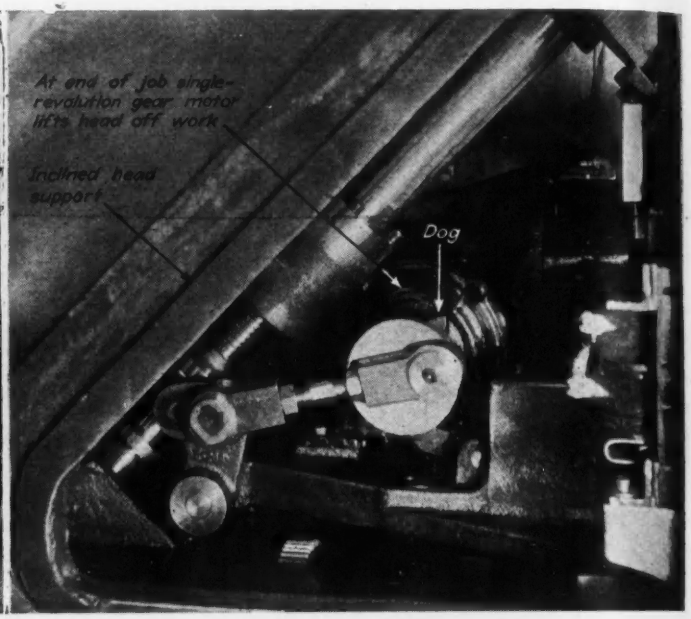
A current-limit adjustment is provided on the control panel to set the maximum current to the value required for each individual application. The armature current is maintained at its maximum safe value and, because

torque is proportional to armature current, maximum safe torque is maintained. In Thy-mo-trol drives which are equipped with field control, the field is not weakened until the armature current starts to diminish, thus assuring that the motor is started with full field.

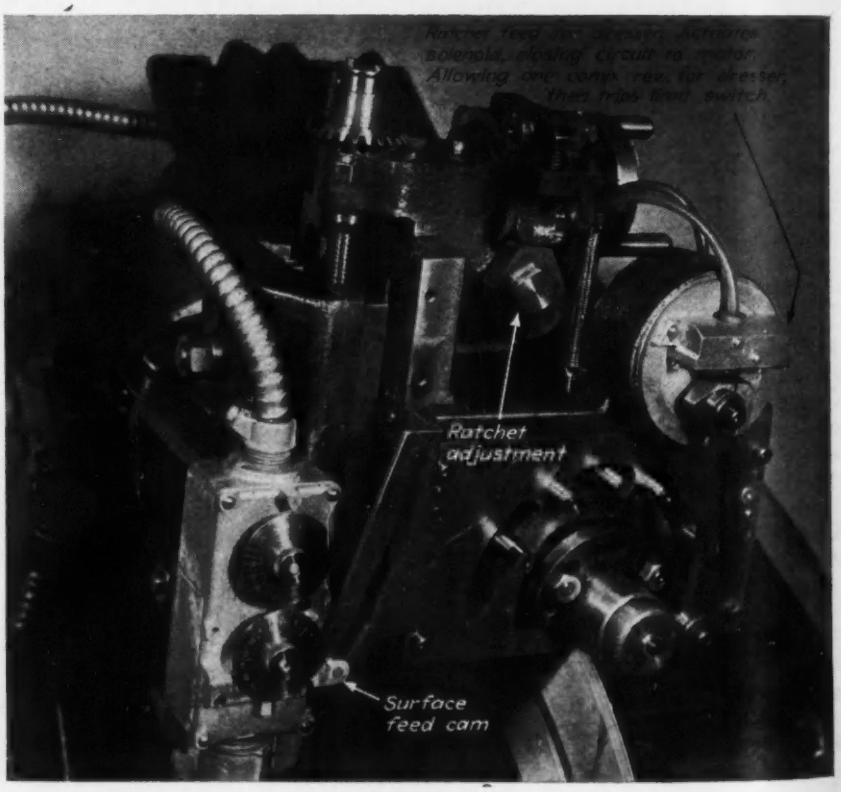
Because acceleration is automatic, the speed adjustment can be preset at any point before starting. Speed of the motor is adjusted by a potentiometer acting through the electronic control circuits, causing the thyratron tubes supplying the armature voltage to increase or decrease their output.

To maintain constant speed, the armature voltage must be increased to overcome the internal armature-voltage drop. The same circuit which measures armature current for current-limit control, automatically increases the voltage output of the armature rectifier as load is applied, and thus, supplies the voltage necessary to maintain an essentially flat speed load characteristic. When operating in the weak field range, if the speed control is changed to reduce speed sharply, the motor momentarily becomes a generator, and its armature whirls with no electrical load and with full field. An abnormally high armature voltage would be generated, were it not for voltage-limiting circuit which automatically limits armature voltage by regulating the field voltage. With a reversing drive a magnetic switch reverses the polarity of the voltage that is supplied to the armature, and allows the drive to operate in either direction.





Work head is mounted on hardened and ground ball bearing ways at 45 deg. angle to assure constant downward pressure on work. Angular mounting improves visibility. Grinding wheel is mounted on a heat-treated alloy steel spindle cantilever mounted in precision-bearings. It is V-belt driven from the 1 hp. d.c. motor. Motor elevation of head, solenoid-controlled, permits rapid clearing of work when indexing from thread to thread and removing work.



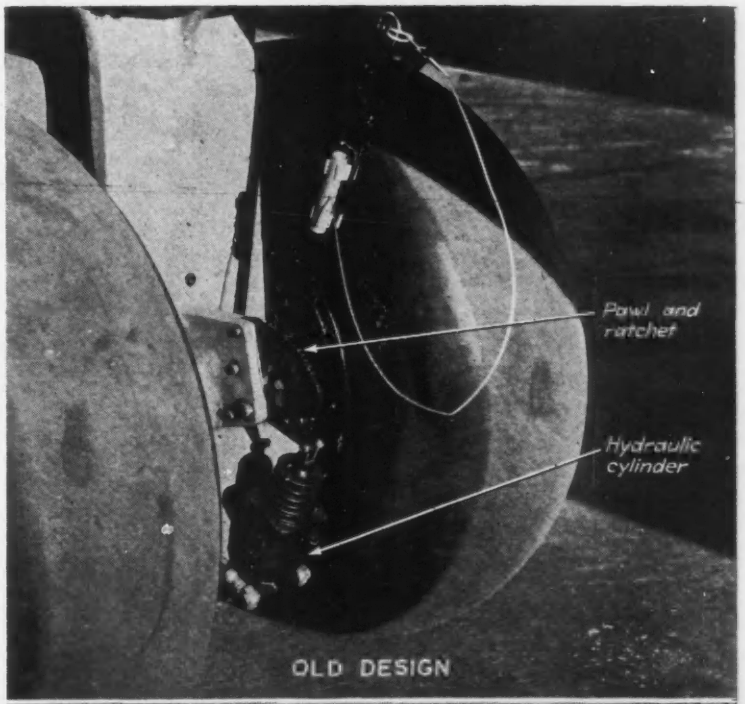
Standard dresser has three dressing diamonds with adjustments provided to dress wheel flanks from 15 to 90 deg. included angle and to dress top of wheel from a flat to an angle of 5 deg. for sloping thread root. Ratchet feed for dresser is solenoid actuated to close circuit to dresser motor. After one complete revolution, a limit switch is tripped and dressing stops.

Redesigned Beaching Gear Brake Provides Greater Safety

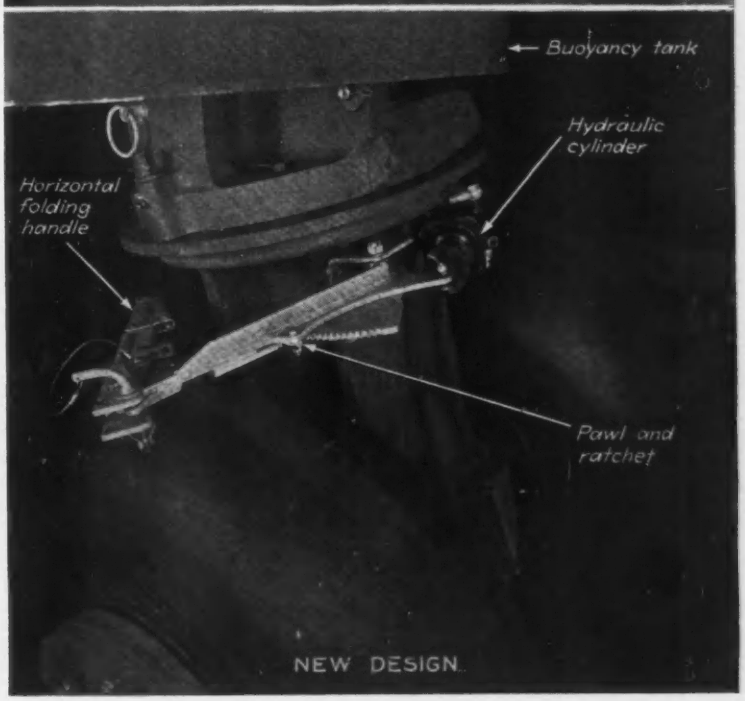
BEACHING GEAR designed by Glenn L. Martin Company consists of heavy welded castings provided with a rectangular buoyancy tank. Moored off shore, the gear floats at an angle with one corner of the tank above water. Gear is attached by two pins at the top to retractable fittings in the hull. These fittings are projected for attachment of the upper strut fitting of the gear. Pins in the lower portion of the gear engage mating socket in the chine of the hull, and a toggle linkage locks the gear in position. A smaller gear, also with two wheels, is similarly attached aft under the tail. This gear swivels completely around to steer the plane as it is beached. When the ship is out of water, this gear is equipped with a 10-foot towing bar.

Beaching gear brake system has been redesigned to provide greater safety. On beaching operations, the plane is towed by cable from a tractor attached to a tow lug under the tail, and hand brakes on the two main gear control the movement of the airplane.

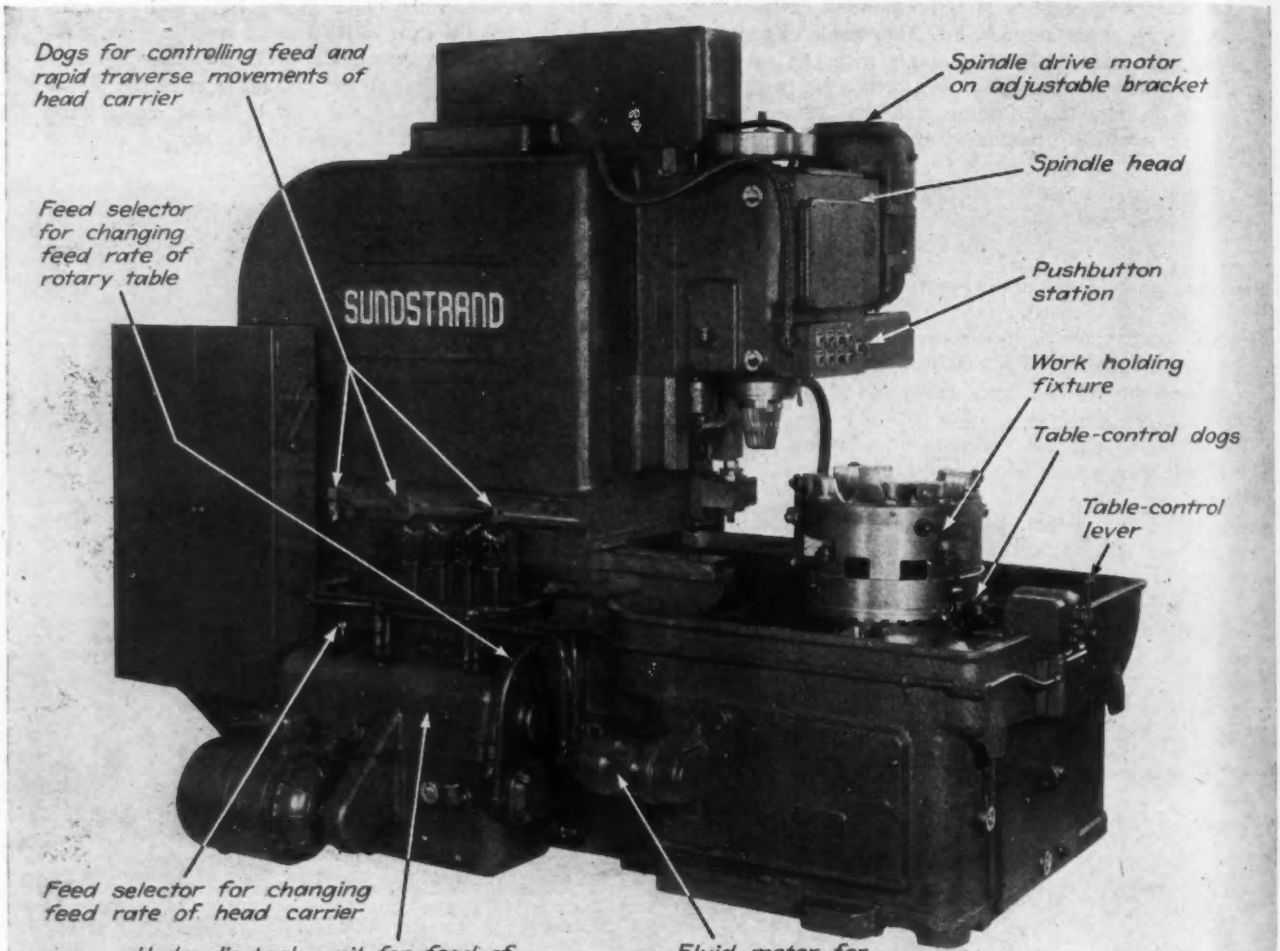
Old brake consisted of a steel handle located between the two wheels of the beaching gear to which was attached a pawl working in a ratchet that maintained brake-on position as hydraulic pressure was applied to the brake drums. The ratchet was held in a lug on the side of the casting, and the bottom of the brake lever was bolted to the hydraulic cylinder rod. The cylinder was located at an angle on another lug at the bottom of the casting. To apply the brake the operator had to walk directly in the path of the wheels as the airplane was being towed from the water.



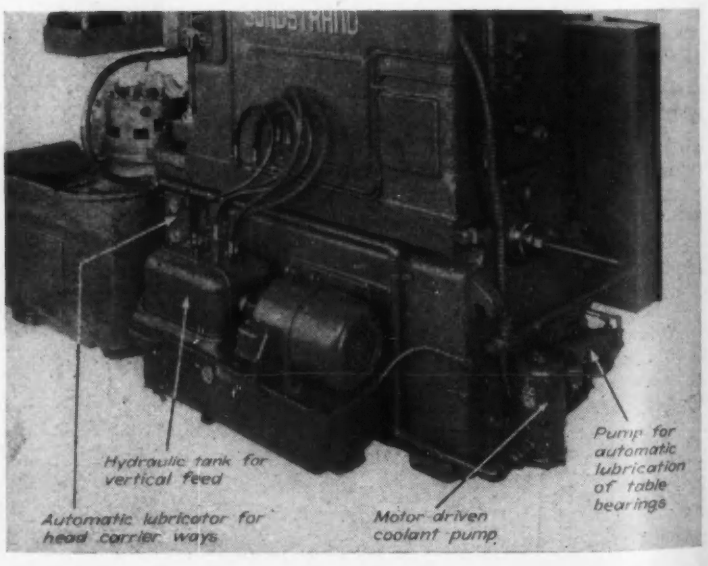
New design, in use for several months, has the brake operating lever in a horizontal position, protruding over the outside wheel, placing the man out of the danger area. The change was accomplished by moving the hydraulic operating mechanism from the bottom to the top of the casting, and redesigning two lugs which are bolted to the casting to accommodate the ratchet and hydraulic cylinder. The brake lever, still made of steel but redesigned for the new horizontal position, is longer and has a hinge for folding when not in use. A phenolic-fiber hand grip, resistant to the action of salt water, substitutes for the wooden grip. The hydraulic cylinder is unchanged except for relocation. The new brake system adds somewhat to the weight of the beaching gear, but in view of the added safety provided this is relatively unimportant, since the gear is aboard the airplane only during transport and ferrying operation.

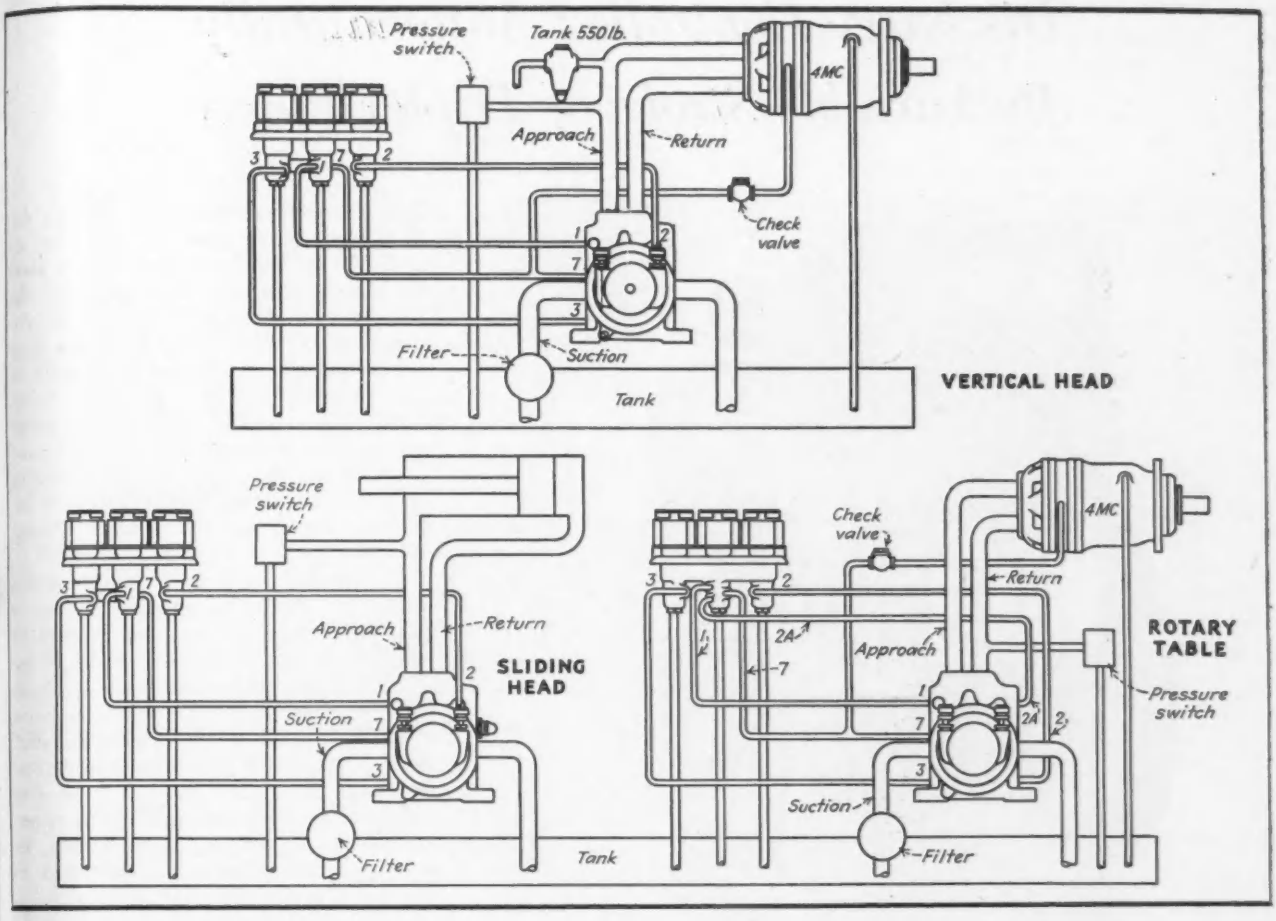


Automatic Cycle Provides Three Interlocked, Independent Motions

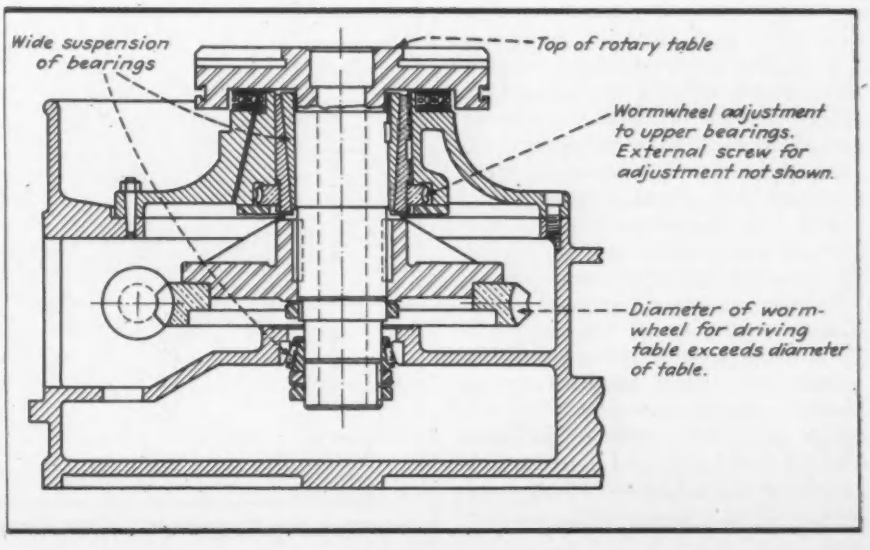

Dogs for controlling feed and rapid traverse movements of head carrier
Spindle drive motor on adjustable bracket
Spindle head
Feed selector for changing feed rate of rotary table
SUNDSTRAND
Pushbutton station
Work holding fixture
Table-control dogs
Table-control lever
Feed selector for changing feed rate of head carrier
Hydraulic tank unit for feed of head carrier and rotary table
Fluid motor for driving rotary table

DESIGNED to form mill the radii and angle on the inside of propeller barrels, this Sundstrand special milling machine design has an automatically controlled hydraulic circuit which provides cross-feed to the column, vertical feed to the head and rotary feed to the table, all timed and interlocked with spindle and coolant pump motors. An angular milling cutter with a radius on the bottom and high-speed steel blades, is mounted directly on the spindle which is driven by a 10 hp. motor. Drive to the spindle is through V-belts with pick-off gears provided to furnish spindle-speed changes in a ratio of 3 to 1. The column is of heavy ribbed construction, carrying the vertical way surface for vertical travel of the spindle head. Provision is made for column to be fed and traversed to and from the workpiece.


Hydraulic tank for vertical feed
Pump for automatic lubrication of table bearings
Automatic lubricator for head carrier ways
Motor driven coolant pump

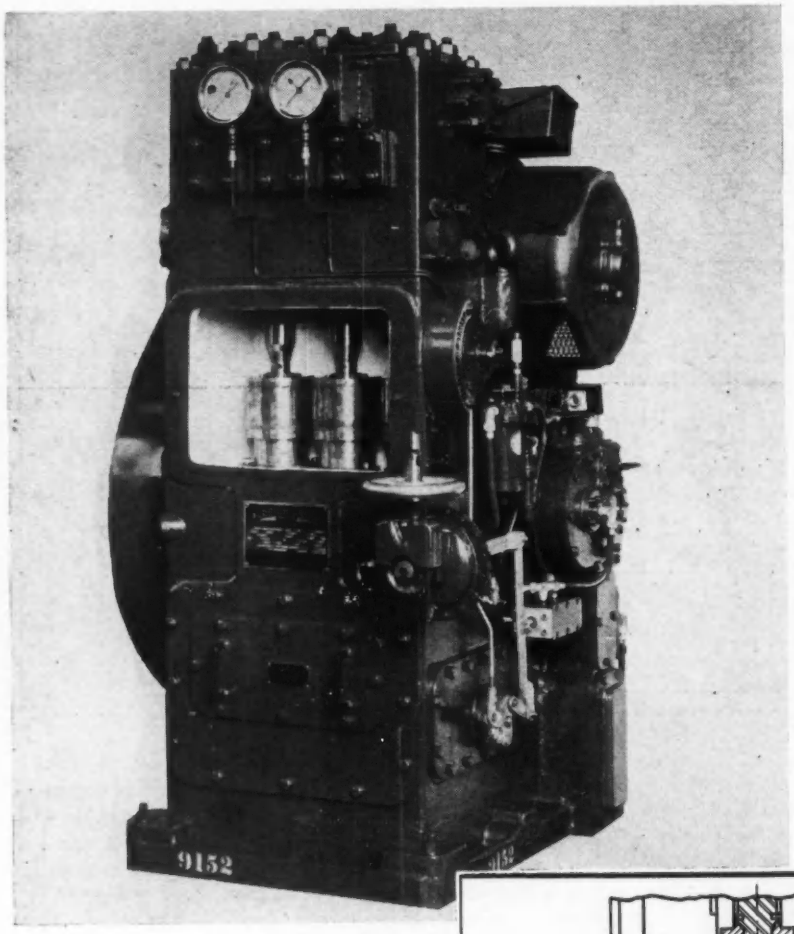


Automatic Operation. Spindle and coolant motors start automatically when the vertical head goes into "feed down" and stop when the head starts up after table-feed cycle has been completed. After the hydraulic and lubricating pump motors are started, the part is loaded and clamped manually in the fixture. Head carrier than rapid traverses crosswise against a positive stop. When the positive stop is reached, the head will start in rapid traverse downward and then automatically change to feed when a dog on the head contacts a limit switch. The head continues to feed downward until a positive stop is reached. Table feed then starts automatically in a counter-clockwise direction and continues through 365 deg. The extra 5 deg. of rotation enables the cutter to clean up at end of cut. Table-control dogs then reverse the direction of table movement so that the table will feed backward or clockwise through approximately 5 deg. against a positive stop. The head then rapid traverses up. At the top of the stroke the spindle carrier rapid traverses back to its starting position. All hydraulic controls are grouped in compact units.


Wide suspension of bearings
Top of rotary table
Wormwheel adjustment to upper bearings. External screw for adjustment not shown.
Diameter of worm-wheel for driving table exceeds diameter of table.

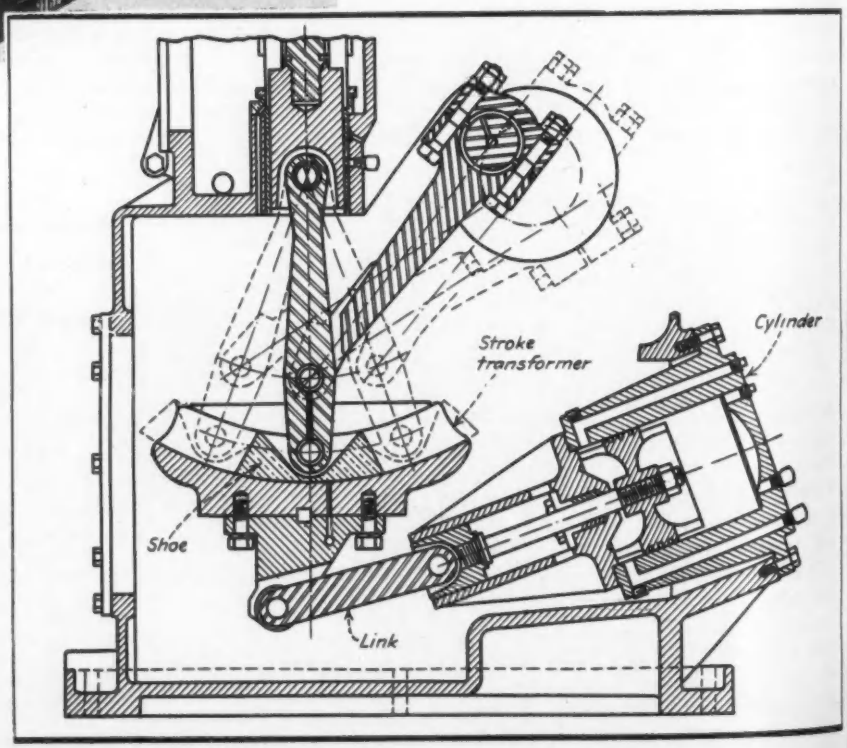
Rotary table, 22 in. in diameter supports the workpiece in a special fixture. The table is of the solid-spindle type. Widely suspended bearings provide rigid support to the table for heavy milling cuts. Upper bearing is adjustable by wormwheel and external screw. Diameter of wormwheel driving table exceeds diameter of rotary table to insure additional stability. This wheel is driven by Sundstrand fluid motor, permitting infinite adjustment of table speeds from ½ in. to 10.1 in. per min.

Discharge Controlled Automatically By Variable-Stroke of Triplex Pump

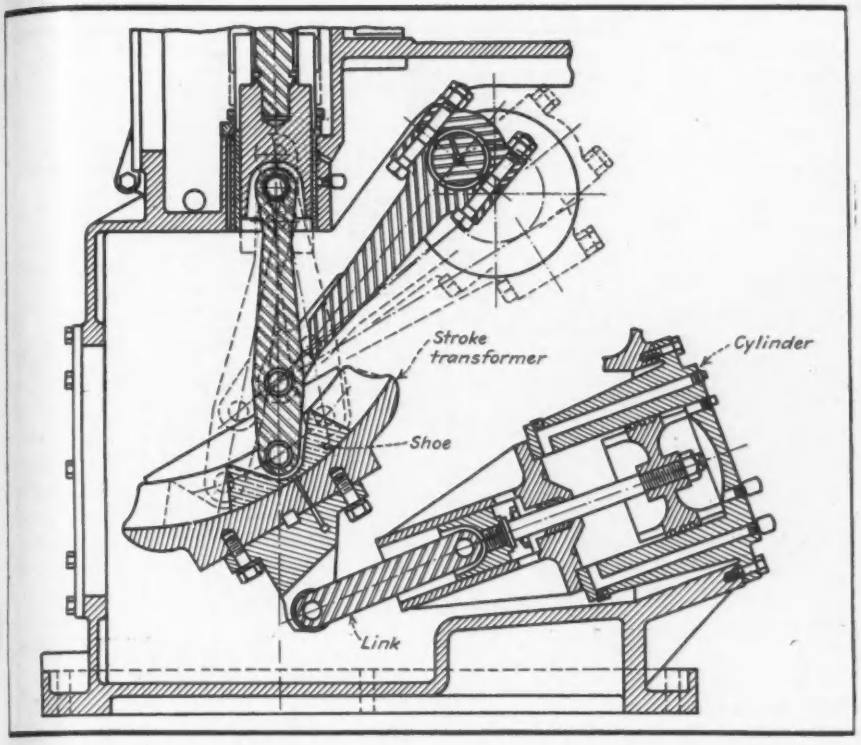


THE Aldrich-Groff controllable discharge triplex pump for pressures up to 15,000 lb. per sq. in. embodies a novel mechanism whereby the stroke of the plunger can be varied to give from zero to maximum rate of output while the pump is running at constant speed. Designed in sizes ranging from 10 to 100 brake horsepower, these pumps are used for boiler feeding, desuperheater feeding, butane and gasoline line pumping, and for furnishing hydraulic power for the operation of extrusion presses and similar hydraulic equipment.

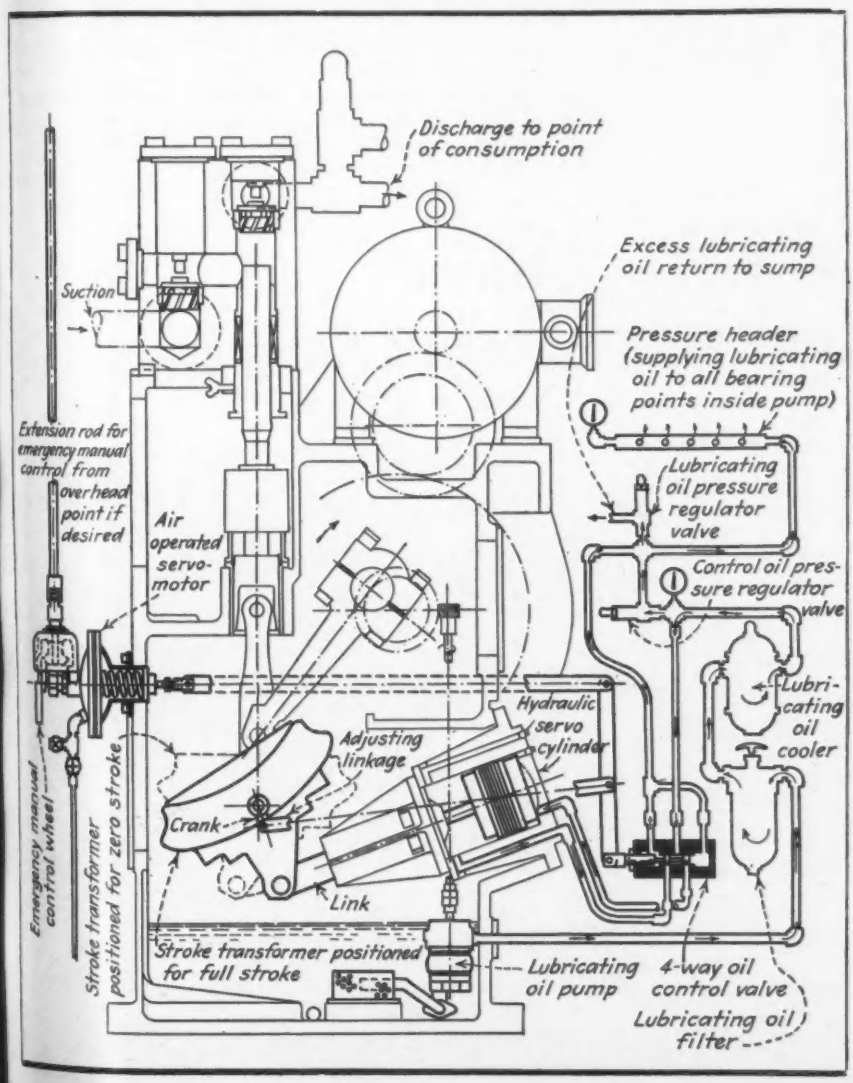
Because the pump can be started at zero stroke at which the starting torque is practically zero, synchronous or squirrel cage motors having normal starting torque characteristics or steam turbines can be used to drive it. Also because the pump runs continuously at constant speed, rather than start-and-stop operation as in the conventional pump, and because there is no bypass of excess quantities of pump output, savings in power are considerable. Its power consumption is proportioned to pump output hence the power curve is smooth without sudden jumps from no load to full load.



Zero - stroke position. Triplex crankshaft rotates at constant speed for all conditions of operation. A connecting rod from each of the three crankpins, spaced 120 deg. apart, oscillates the driving link. In the position shown, rotation of the crankshaft merely oscillates each of the links about its respective cross-head pin as a center, which is also the instantaneous center of curvature of the circularly curved track of the cast semi-steel stroke transformer in which the oscillating guide shoe travels. The more the stroke transformer is tilted to its maximum of 30 deg., the greater the stroke imparted to the plunger. Design provides a variable stroke mechanism lacking the usual multiple joint-connected floating linkage that does not lend itself to the transmission of heavy reciprocating loads.



Full-stroke position. Here the transformer has been tilted to an angle of 30 deg. giving the plungers the maximum stroke and hence producing maximum rated delivery of the pump. The stroke transformer can be set to any intermediate position, adjustment being by means of a hydraulic servo-piston controlled by a 4-way valve, operation of which is regulated by an air-operated diaphragm servo-motor. Any other means for adjusting the position of the stroke transformer can be used. The relative positions of transformer and connecting rod are such that the pull of the connecting rod on the up-stroke of the plunger relieves the downward push on the guide shoe exerted by the plunger. A mechanically driven pump furnishes lubrication at 20 lb. per sq. in. to all bearing surfaces. The same pump furnishes the oil for operating the hydraulic cylinder.


Discharge to point of consumption
Excess lubricating oil return to sump
Suction
Pressure header (supplying lubricating oil to all bearing points inside pump)
Extension rod for emergency manual control from overhead point if desired
Lubricating oil pressure regulator valve
Air operated servo-motor
Control oil pressure regulator valve
Lubricating oil cooler
Hydraulic servo cylinder
Adjusting linkage
Crank
Link
Emergency manual control wheel
Stroke transformer positioned for zero stroke
Stroke transformer positioned for full stroke
Lubricating oil pump
4-way oil control valve
Lubricating oil filter

Automatic operation. In the design shown here, if a greater pump discharge is called for, pressure acts upon the diaphragm of the spring loaded pneumatic servo-motor causing the hydraulic valve operating link to move to the left. Hence the piston of the 4-way valve moves to the right of its central "on lap" position shown. This opens the path for the flow of hydraulic oil, at about 125 lb. per sq. in. pressure to the outer end of the hydraulic servo cylinder causing the piston to move inward. The oil discharged from the hydraulic cylinder is forced through a relief valve set at about 25 lb. per sq. in. and into the lubricating oil system. Thus, the bearing surfaces receive the same amount of lubricating oil, regardless of the operation of the hydraulic cylinder. Direction of flow through the various oil lines during this operation is indicated by the arrows.

The piston, connected to the stroke transformer by a link, continues moving inward until the transformer has been tilted sufficiently to cause the follow-up valve control linkage to restore the valve piston to its "on lap" position. Operation of the pump then continues at this intermediate stroke and delivery rate until more or less pump delivery is demanded by the increased or reduced pressure of the control air. The control air pressure is regulated by a float switch or similar device. The fac-

tor controlling the rate of discharge of the pump may be the level of the liquid, as in a boiler or tank, the pressure, the temperature or any other variable in the system that predicates the rate of discharge desired of the pump.

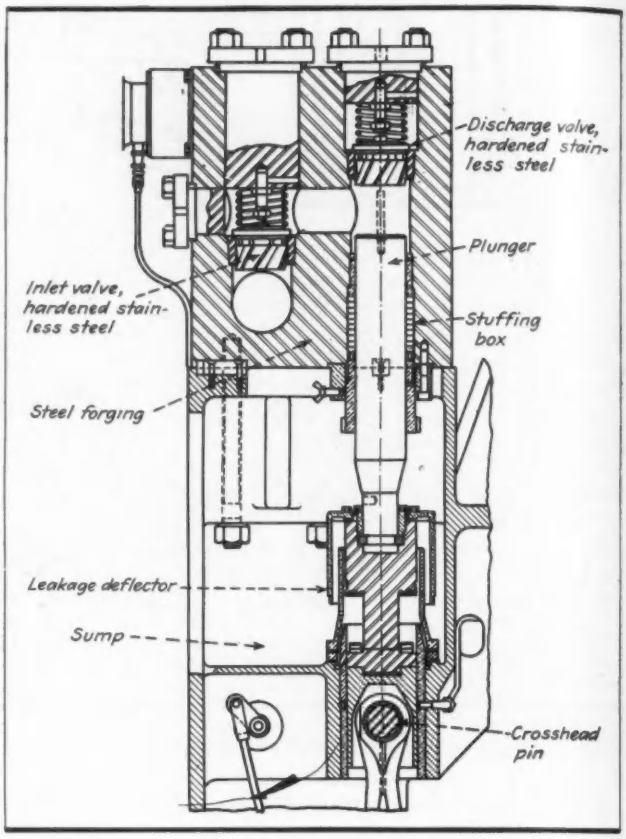
Special design features. The fluid end of the pump is at the top of the unit making possible a compact design of pumping chamber having minimum clearance volume and highest volumetric efficiency. Vertical design also gives best access to plungers and valves and relieves the stuffing box packing of having to help support the plunger, as in the case of horizontal design.

Pump cylinder block is machined out of a single steel forging thus avoiding manifold joints. Forging also eliminates possible leakage on account of metal porosity.

Suction and discharge valves, valve seats and springs are interchangeable. Valves and seats are of hardened stainless steel.

Plungers are connected to their respective cross-heads by means of a split ring and a retainer nut which constitute a "removable flange." This construction allows the plungers to freely center themselves within the stuffing box packing and permits removal of the plunger upward and outward through the discharge valve seat opening without disturbing any other mechanism.

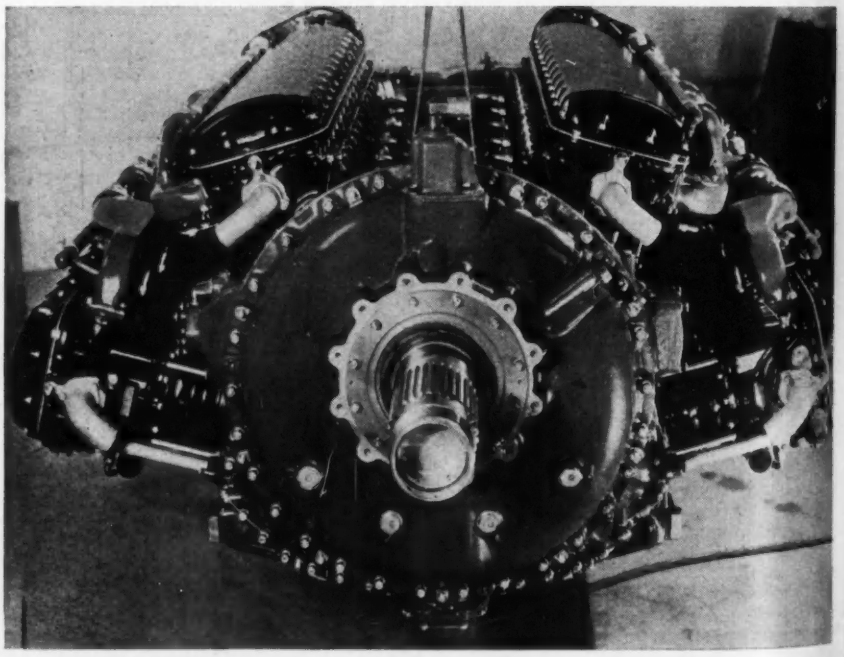
Leakage deflectors prevent any fluid being pumped from leaking into the pump lubrication system and likewise prevent any lubricating oil from working upward to the pumping plungers and thereby contaminating the fluid being pumped. This is important in boiler feedng where the presence of oil in the water is dangerous to the heating surfaces.



New W-Type Allison Rated World's Most Powerful Engine

FOUR BANKS of six cylinders each make up the new W-type, liquid-cooled Allison engine which has maximum ratings approximating 3,000 hp. The engine has successfully passed Wright Field tests. The new engine has exactly twice the piston displacement of the present Allison 12, but through design simplification 97 percent of the parts are interchangeable with present Allison engines. Thus spares and replacements are already available in air force depots and little or no additional training will be required for maintenance men.

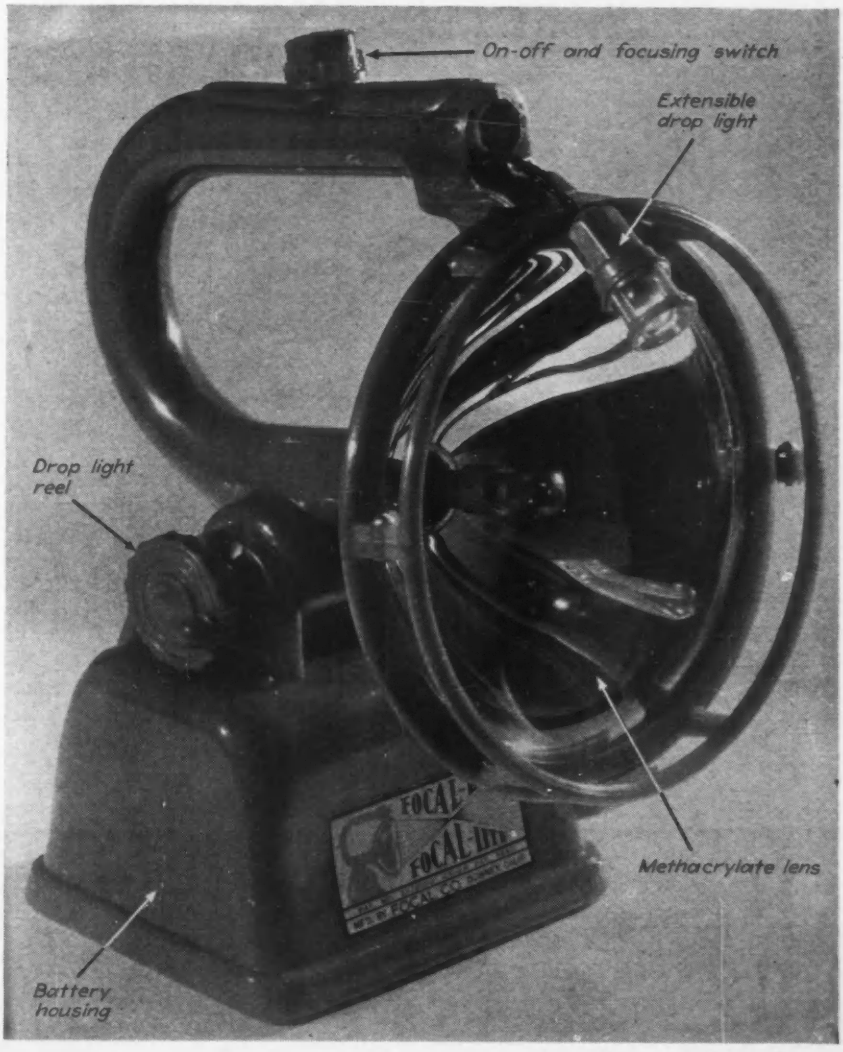
Weight per hp. has been decreased below that of the 12-cylinder series, which was less than one lb. per hp. Despite the large increase in horsepower, frontal area has not been increased proportionately, hence the design of the engine lends itself to "buried" installation and consequent reduction of power-plant drag.

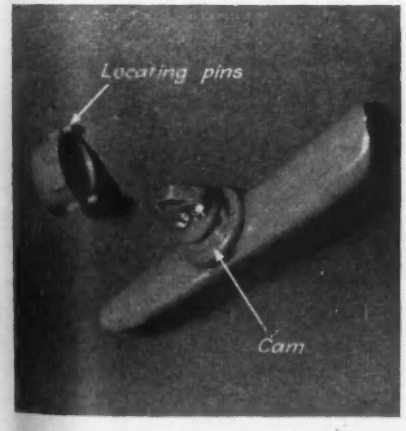


Plastic Hand Lantern Operates on Flashlight Cells

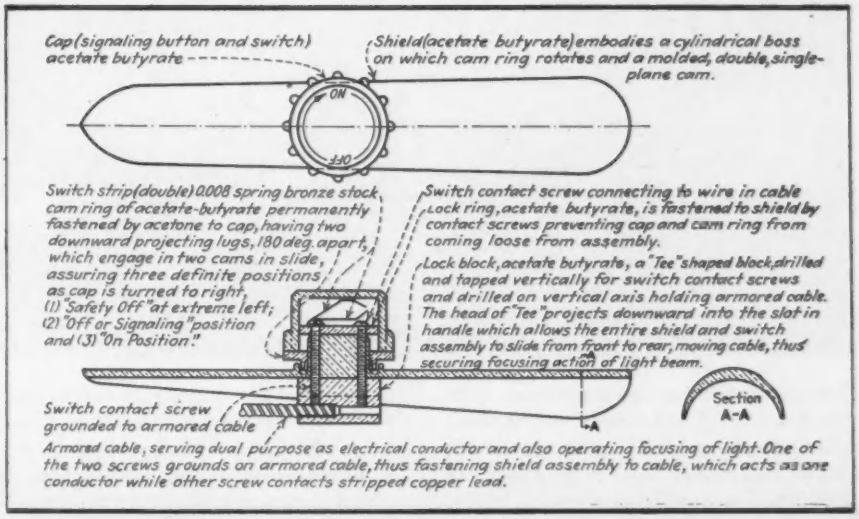
NEW and improved model of the "Focal-lite" hand light, first described in PRODUCT ENGINEERING, June 1942, page 315, has a formed acrylic plastic lens and a molded acetate control-switch assembly which replaces a die-casting. Battery mount is also redesigned. Other features, which include lightweight butyrate housing, retractable drop light, and ease of focusing are retained. New model operates on flashlight batteries.

Battery mount, made of vulcanized fiber and extruded cellulose acetate-butyrate tubing, accommodates six flashlight cells. Movable contact arms on top and bottom plates provide settings of 3, 4½, 6, 7½ and 9 volts. Battery mount is fitted into molded-in recess at base of lamp. It is accessible by removal of base cover plate, fastened with two screws.





Switch is entirely of plastic except for contacts. Former die-cast assembly had a removable cap. In new design, switch is permanently assembled by acetone cement. Assembly is not subject to corrosion. Cam ring is permanently fastened to cap, which has two downward projecting lugs to engage the cams, assuring three definite operating positions for the switch.


Cap (signaling button and switch) acetate butyrate
Shield (acetate butyrate) embodies a cylindrical boss on which cam ring rotates and a molded, double, single-plane cam.
ON
OFF
Switch strip (double) 0.008 spring bronze stock, cam ring of acetate-butyrate permanently fastened by acetone to cap, having two downward projecting lugs, 180 deg. apart, which engage in two cams in slide, assuring three definite positions as cap is turned to right, (1) "Safety Off" at extreme left; (2) "Off or Signaling" position and (3) "On Position."
Switch contact screw connecting to wire in cable
Lock ring, acetate butyrate, is fastened to shield by contact screws preventing cap and cam ring from coming loose from assembly.
Lock block, acetate butyrate, a "Tee" shaped block, drilled and tapped vertically for switch contact screws and drilled on vertical axis holding armored cable. The head of "Tee" projects downward into the slot in handle which allows the entire shield and switch assembly to slide from front to rear, moving cable, thus securing focusing action of light beam.
Switch contact screw grounded to armored cable
Section A-A
Armored cable, serving dual purpose as electrical conductor and also operating focusing of light. One of the two screws grounds on armored cable, thus fastening shield assembly to cable, which acts as one conductor while other screw contacts stripped copper lead.

WARTIME DESIGN NEWS

Fighter-Bombers Emphasized

Production of long-range fighter-bombers, which are now carrying bomb loads somewhere near those of medium bombers but which retain a fighting flexibility not possessed by the heavier craft, is receiving increasing emphasis in the Army's aircraft program. The fact that the latest model of Republic's P-47 Thunderbolt can carry two 1,000-lb. bombs slung under its wings illustrates how fighters are moving in on the field of the medium bomber. On the other end of the scale, with the B-17 and B-24 heavy bombers considered "the last of the small bombers," the mediums will have even less application. North American will halt production of B-25 medium bombers at its Inglewood, Cal., plant this month and concentrate on production of P-51 Mustangs, another highly successful fighter bomber.

The need for long-range escort fighters which can go all the way with the heavies helps dictate the above strategy. The B-26 Marauder, another medium bomber, is also being tapered off in favor of extreme emphasis on the super-heavy bombers, long-range fighters and fighter-bombers and new secret types of aircraft. Experts emphasize that the medium bombers will not be discarded, but merely played down so other ships in greater demand can be pushed ahead.

More Plane Refinements

The new Thunderbolt has been fitted with a plastic full-view canopy similar to that of the P-51, which is electrically operated. In addition, the water-injection device (PRODUCT ENGINEERING, June, 1944, page 378) is standard equipment, adding several hundred horsepower in spurts. Internal fuel capacity has been increased to give the plane double its early cruising range, and new paddle propellors have added 400 ft. per min. to its rate of climb.

The Thunderbolt is only one of five types of Army fighter planes equipped to fire rockets from under its wings. Others are the P-38 Lightning, the P-40 Warhawk, the P-39 Airacobra and the P-51 Mustang.

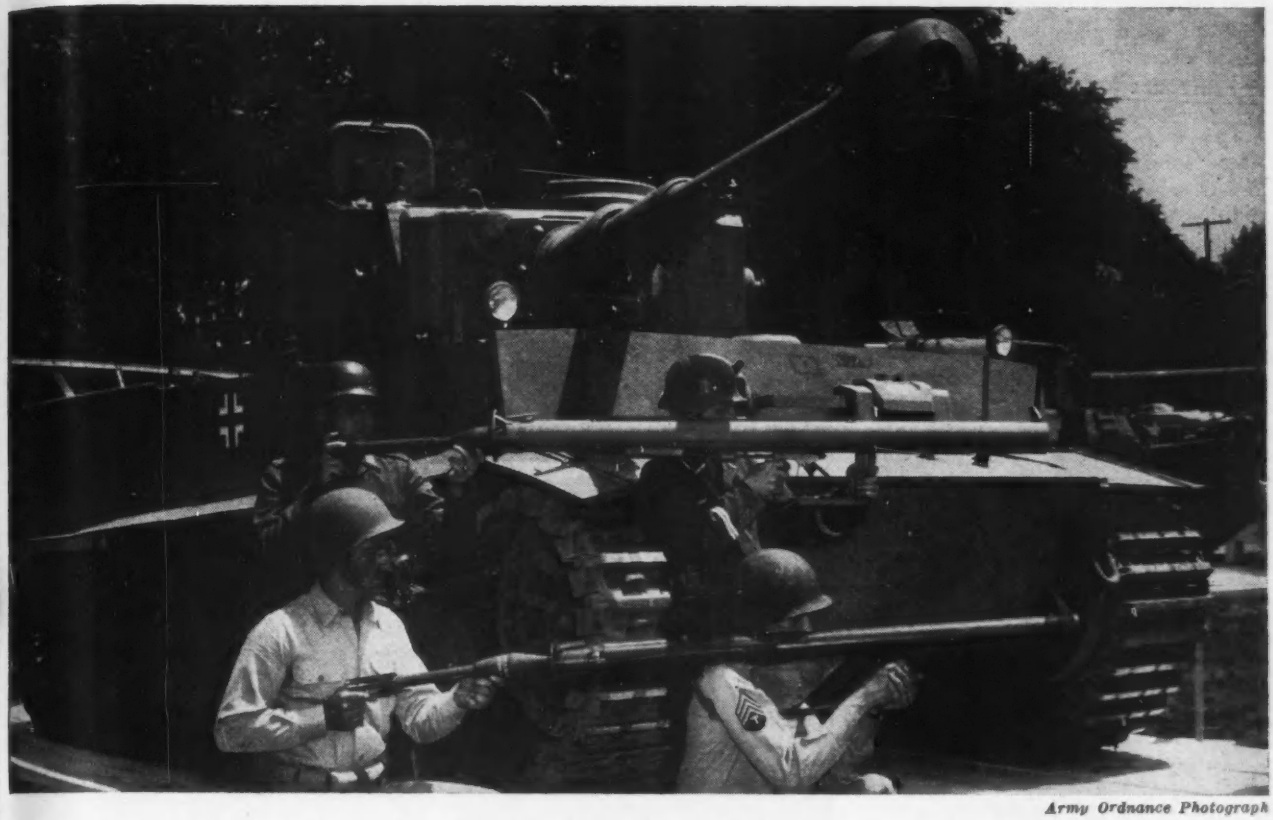
The English, too, are making constant improvements and refinements. Among the latest to be announced are two planes with wing design so unorthodox that they give the impression of flying backwards. One is a shipboard fighter, the other a twin-engine bomber, both of which have been tested successfully. The fighter appears to have been built in reverse, with the sharp end of the fuselage forward and the stubby portion to the rear. Aft of the cockpit is a small high wing, while in the normal location of the tail surface is a low wing of normal size for such plane. Atop this wing are vertical stabilizers and rudders. A tail wheel has been added to the normal tricycle landing gear. This feature, plus the extreme visibility afforded by the small wing forward, makes the plane particularly suitable for shipboard operation.

German Secret Weapon Captured at Anzio

Signal Corps Photo

Nazi one-man submarine captured at the Anzio beachhead in Italy when its 17-year-old pilot grounded there. Top is the chamber in which the pilot lies. Beneath is the torpedo, which is released in the vicinity of the target, after which the pilot theoretically returns to his base. The British were recently reported to have developed a two-man submarine with a torpedo similarly suspended. The crew was supposed to have donned diving suits and straddled the chamber and torpedo, diving in the vicinity of the target to fire the torpedo at close range.

U. S. Bazooka Superior to German Rocket Gun and Tank



Army Ordnance Photograph

Master of the Nazi "bazooka" and "Tiger" tank shown here is the redesigned U. S. bazooka demonstrated by the two-man team in the foreground. This demonstration was part of the Army Service Forces' "Weapons of War" show held recently at Washington. The German "bazooka" is of larger caliber than the U. S. weapon, 8.8 cm. against 2.36 in., and carries a larger penetrating charge but it is less effective than our weapon. The German rocket gun has blown up often in service, and captured German Army orders show that it is to be fired only on direct order of an officer. The German model is almost an exact copy of the first U. S. bazooka except in the caliber, the handles and placement of the firing mechanism. The new U. S. "bazooka" mounts a screen just behind the muzzle to break up any particles which might be hurled back by the projectile. It has only one pistol grip rather than the original two and carries a sling to improve marksmanship. The Tiger tank in the background is a 60-ton vehicle. Its type has been knocked out by bazooka fire. Note the muzzle brake on the gun, which is believed to be an 88-mm. cannon.

Germans Claim Jet Planes

Nazi propagandists claim the Luftwaffe has two jet-propelled planes sufficiently far along for operational use, but neither has yet been encountered by Allied airmen. A new Heinkel bomber, said to carry four engines mounted in tandem driving two propellors, also is reported. This craft is further reported to use means of assisting its takeoff and to be fitted with rockets for rapid acceleration when necessary. A five-engine Heinkel 111-Z was encountered towing gliders over France.

Whether or not the Germans have any jet-propelled planes ready for operational work, they have two new fighter models, a Focke-Wulf 290 and a Messerschmitt 209, the latter endowed with almost miraculous climbing ability, according to Mustang pilots who encountered the plane. The new Focke-Wulf has an elongated nose which indicates the use of a different power plant; the FW-190 used a radial engine.

Substitutions Prove Out

Col. G. S. Brady of Army Ordnance recently disclosed that at least half of the material substitutions which have been made in war goods so far have been real improvements and not merely makeshifts. In some instances, he said, substitute materials proved better for the job in question than the materials they replaced. In others, he said, the substitutes forced such careful redesign and reconsideration of the entire problem as to yield improvements. When some substitutions are replaced by original materials, he went on, better use will be made of the standard material through knowledge brought out in redesign to accommodate a substitute.

"Raincoat" for Planes

Army Air Forces Service Command has come up with a quick new method of preparing aircraft for exposed overwater shipment. Liquid plastic, sprayed over the entire plane, coagulates within 30 minutes into a thin, water-resistant film. It will withstand temperatures from 180 deg. above to 30 deg. below zero, an Army officer revealed, and can be peeled off in half an hour by several men, with the peelings shipped home and reused.

Glass Tank Panels

Allied tank commanders henceforth will get a better view of any action in which they are engaged by means of high-strength glass panels so mounted on tanks as to afford greatest possible vision with the hatch closed. Made of laminated plate glass by Libby-Owens-

Ford, the panels are known as prismatic viewing blocks, and utilize for the first time the refractive properties of bullet-resistant plate glass to obtain a periscopic effect. The panels are resistant to all high-velocity projectiles except armor-piercing shell.

The War Department has also released information disclosing that the M-4 General Sherman tank mounts, in addition to its 75mm. gun and several machine guns, a 2-in. mortar for projecting smoke bombs and various types of grenades.

From Sweden comes news of a new British mine which can be dropped into the sea without a parachute, apparently set to remain at any predetermined depth. The mine is detonated by the sound waves emanating from passing ships, the report adds. The new British development surpasses the earlier acoustical mine in that it will be undisturbed by several ships passing near it but will explode when a fourth or fifth ship goes by. The report, credited to a Swedish merchant sailor and thus open to the charge of being propaganda, also claims that the mine is proof against normal sweeping operations.

Port Repair Ships

Designed to handle virtually any contingency which may be faced when a harbor is wrested from the enemy, the first of a small fleet of Army port repair ships was shown recently after a successful shakedown cruise. Facilities of the ship, a converted freighter whose gross tonnage is, with port repair equipment, about 2,500, include equipment for divers who might have to clear away underwater debris, five booms with which to clear away dock obstructions or handle similar heavy loads, fire-fighting equipment and a complete machine shop. A cat-head on the bow can handle 40-ton loads. The five foremast booms include four ten-ton booms and one of 30-ton capacity, each equipped with a power winch. A dismountable pontoon outfit provides 50-ton buoyancy. Welding and electrical repair shops have power units sufficient to supply the ship.

New Devices Needed

The National Inventor's Council is seeking new devices whose need has developed with the change in military operations and requirements.

Typical of the problems which the armed forces are now seeking to solve are the following:

A durable coating for glass surfaces to reduce glare. It must be capable of withstanding field conditions.

An optical method of distinguishing between an artificial green and a natural green.

A gas mask which would enable the wearer to speak normally and be heard clearly.

Protection against flame-throwers.

A life vest which will inflate automatically and turn an unconscious man on his back when he is thrown overboard by concussion.

Fire-fighting equipment sufficiently effective to control fires in fighting tanks long enough to allow evacuation of personnel. The device must not be injurious to the personnel and should be manually controlled and operated.

Suggestions which have been received by the council and translated into fighting equipment are now in action, including a new electrical firing device for the bazooka which replaces the batteries originally carried in the stock. This trigger-actuated mechanism creates an electrical spark formerly furnished by the batteries.

Likewise, the new signalling mirror, standard equipment on all life rafts, evolved from the inventors council. When kapok became scarce, it was a suggestion to the council that milkweed floss be used which enabled continued manufacture of life preservers and similar buoyant clothing, insulation and soundproofing.

Anti-Aircraft Gun Hits Planes in Stratosphere

U.S. Army Photo

The Army's "stratosphere" 120-mm., 4.7-in. anti-aircraft gun can fire a shell 20,000 ft. higher than any other known anti-aircraft weapon. It may be the weapon referred to a year ago by Army Ordnance officers who told of a new anti-aircraft gun accurate at 60,000 ft. This weapon can be dropped rapidly onto its outriggers from its wheels for firing and picked up again quickly by the 15-ton hydraulic jacks on its outriggers. Gunners to the right are holding the shells, those to the left the powder cases. An automatic fuse setter cuts the desired fuse and a loading mechanism receives shell and charge, loads the gun and closes the breech without further action by the crew.

Product Design At Taxpayers' Expense

UNDER the title "Slave or Free," Dr. Herbert K. England in his book "Listen My Children," tells the story of the skylark who yielded to this tempting song of a little old peddler:

> Who will buy? Who will buy?
> I am selling in all weathers
> Fine and fat juicy worms
> In exchange for skylark feathers.

Here was a chance to get something for nothing, at least practically nothing because the skylark had plenty of feathers. So the skylark bought two worms for a feather and spent the remainder of the day loafing and chirping.

The first time the skylark did this, it hurt his conscience a little. By the next day he forgot his qualms and again exchanged a feather for juicy worms. After that it was easy. But soon the skylark couldn't fly so high and finally he couldn't fly at all. He died a miserable death.

Perhaps never before in world history have the peoples of a nation received so many offers of "juicy worms" in exchange for "skylark feathers" as have the people of the U.S.A. during the past decade. A special "juicy worm" now being prepared is one bred by the Smaller War Plants Corporation, a wartime government bureau. The scheme is set forth in bill S. 1913 which at this writing has been referred to the Senate Banking and Currency Committee. Under the bill's terms, the S.W.P.C. would be known as the Small Business Corporation and its operation would be projected into the anticipated peace years ahead.

The Corporation would offer its aid in perfecting inventions and in developing the product designs of small manufacturers (500 or fewer employees), and individuals, at the taxpayers' expense. Indeed, this bureau would loan taxpayers' money to the enthusiastic inventor or little manufacturer to stake his business enterprise.

Naturally, the government's bureaucrats want a few "skylark feathers" in return. The resulting patents would be the property of the government bureau. The little fellow who applied for the development of the design would be given a free license to manufacture it. So would his small-sized competitors, that is, if the Chairman of the S.B.C. decided that way. And if it happens that the politically appointed Chairman has a particularly friendly feeling toward some big company, he might arrange to let that firm have a license too. There is nothing in the law to prevent it.

To carry out the purposes of this act there would be established, "a field consulting service . . . of persons experienced and trained in the problems of small business and capable of counseling owners and managers of small business in management and other problems . . ." No estimate of the cost of operating the S.B.C. is given but the capitalization is stated as One Billion Dollars, all of it taxpayers' money.

Examination of this proposed bill shows that the proposed Small Business Corporation would be granted the authority "to require such information and data from applicant concerns as it shall deem necessary to carry out the purposes and provisions of this Act." The bill proposes further that Congress give the Corporation power to make its own rules. In the light of experiences with other bureaucratic agencies, these "rules" would have the force of law.

Are the American people going to keep on swapping the feathers of their wings of freedom for juicy worms offered by power-seeking politicians? They will if they don't rise in courageous opposition to proposed bureaucratic measures such as S. 1913!

G. F. Voidenholt

Calculation of Flow Velocity From Fluid-Flow Friction Diagram

R. C. BINDER
School of Mechanical and Aeronautical Engineering, Purdue University

Additional coordinates on the well-established friction diagram make possible a direct calculation of the flow velocity in pipes and fluid couplings. Use of this method enables one to determine directly the velocity factor needed for calculating the torque transmitted through a fluid coupling, and the power at each shaft.

IN MANY problems dealing with the flow of fluids in which the head H, the length L and diameter D of the pipe are known it is not possible to find the dimensionless friction factor f and the velocity V directly from a plot of the friction factor f versus the Reynolds number R alone. Certain lines and coordinates, however, can be added to the friction diagram that make possible a direct determination of the flow velocity.

It is common practice to express the friction loss H for pipe flow in the form

$$H = f\left(\frac{L}{D}\right)\frac{V^2}{2g} \qquad (1)$$

where f is a dimensionless friction factor, L is the length of path of flow, D is dimeter, V is velocity, and g is the acceleration of gravity. If L is expressed in ft., D in ft., V in ft. per sec., and g in ft. per sec. per sec., then H is in feet.

Many experiments have resulted in a correlation of the friction factor f as a function of a dimensionless parameter R, called Reynolds number,

$$R = wVD/g\mu$$

where w is the specific weight of the fluid in lb. per cu.ft., μ is the absolute or dynamic viscosity of the fluid in lb. sec. per sq.ft. Any set of consistent units can be employed in determining R. A typical plot of some data is shown in Fig. 1. A plot of f versus R is a simple, sound, and convenient method for correlating data for different fluids and conditions.

Note that the ratio $g\mu/w$, in ft. squared per sec. is the so-called "kinematic viscosity" of the fluid, a ratio of dynamic viscosity divided by mass density. Frequently, in dealing with oils particularly, measurements are made with a Saybolt viscometer. The Saybolt reading can be correlated empirically with the kinematic viscosity. An empirical correlation between Saybolt Seconds Universal and kinematic viscosity, based upon experimental results, can be found in the A.S.T.M. Standards, 1939, Part III, page 215.

Rearrangement of Equation (1) and the expression for Reynolds number gives the form

$$\frac{2Hw^2D^3}{Lg\mu^2} = fR^2 \qquad (2)$$

For practical purposes, to get smaller numbers, it is convenient to take the square root of each side of Equation (2) to give,

$$\frac{W}{\mu}\sqrt{\frac{2HD^3}{Lg}} = R\sqrt{f} \qquad (3)$$

Each side of Equation (3) is a dimensionless ratio; any set of consistent units can be used in calculating values. The left side of Equation (3) has known factors, whereas the right side has unknown factors which are coordinates of the friction diagram.

Let

$$B = \frac{w}{\mu}\sqrt{\frac{2HD^3}{Lg}} = R\sqrt{f} \qquad (4)$$

If constant B lines are added to a diagram of f versus R, then both f and R can be determined if B is known for a particular application.

Lines of constant B values have been added to Fig. 1. Note that lines of constant B are simply straight lines on the log-log plot. Taking the logarithm of each side of Equation (4) gives

$$\log B = \log R + \frac{1}{2}\log f$$

which is the equation of a straight line if logarithmic coordinates are used.

The method of determining the flow velocity V directly is illustrated in Fig. 2. The B number is calculated first. The intersection of the B line with the curve on the friction diagram gives point A. Thus R is determined, and the velocity V can be calculated from

$$V = \frac{Rg\mu}{Dw}$$

The data in Fig. 2 were taken, more or less as samples, from well-known pipe experiments. Any other pertinent data can be plotted and B lines constructed.

One of the major steps in predicting the performance of a fluid coupling as outlined in "A Simple Method for Calculating Hydraulic Coupling Performance" by R. E. Bruckner, PRODUCT ENGINEERING, December, 1942, page 698, involved a trial and error calculation of the velocity of flow in the coupling.

A fluid coupling as shown in Fig. 3 consists essentially of a centrifugal pump impeller and a turbine runner. In operation, the pump impeller rotates at a higher speed than the turbine runner. It is the difference in centrifugal force resulting from this difference in speed that causes circulation of the fluid, usually oil, between the two members. For a state of steady motion, a power balance will show that the power resulting from the centrifugal force action equals the friction loss in the vane passages.

A method for calculating the centrifugal head H has been outlined in Bruckner's article. The slug of fluid in each rotating member gives rise to a certain pressure which is caused by centrifugal action. The difference in pressures divided by the specific weight w of the fluid gives the head H, commonly expressed as energy per unit weight of fluid flowing, as ft. lb. per lb., or simply feet. For water, w is 62.4 lb. per cu. ft. For most hydraulic coupling oils, w is about 56 lb. per cu.ft. If H is known, then the friction loss is H when the motion is steady.

Given the major dimensions of a coupling and the speed of each shaft, the general problem is to determine the torque transmitted through the coupling. The torque depends upon the velocity of flow. The friction loss depends upon the velocity. It was suggested in R. E. Bruckner's article that a velocity first be assumed, then the loss calculated, and this procedure continued until the proper velocity was discovered.

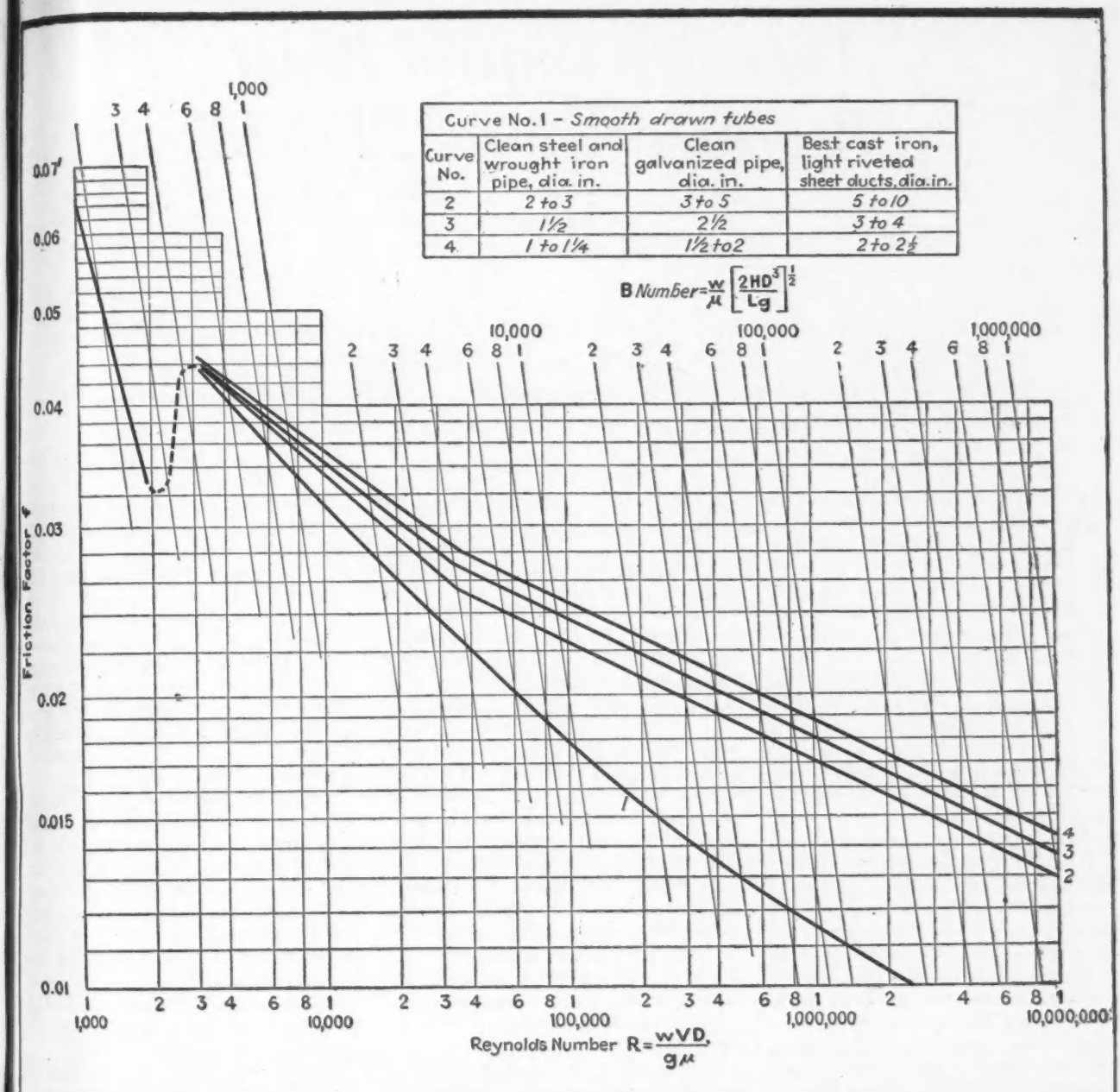

Curve No. 1 – Smooth drawn tubes
Curve No. | Clean steel and wrought iron pipe, dia. in. | Clean galvanized pipe, dia. in. | Best cast iron, light riveted sheet ducts, dia. in.
2 | 2 to 3 | 3 to 5 | 5 to 10
3 | 1½ | 2½ | 3 to 4
4 | 1 to 1¼ | 1½ to 2 | 2 to 2½
B Number = $\frac{w}{\mu}\left[\frac{2HD^3}{Lg}\right]^{\frac{1}{2}}$
Friction Factor f
Reynolds Number R = $\frac{wVD}{g\mu}$
1,000
10,000
100,000
1,000,000
10,000,000
0.07
0.06
0.05
0.04
0.03
0.02
0.015
0.01
4
3
2

Fig. 1—Relation between friction factor and Reynolds number for flow in pipes. Supplemental coordinates for B numbers have been added. Fig. 2—Method for determining flow velocity directly from friction diagram with the aid of additional coordinate. Fig. 3—Sectional view of a fluid coupling.

In the fluid coupling problem, H, L, and D are known. But f and V are not known, and cannot be found directly from a plot of f versus R alone.

The trial and error calculations for determining the velocity of flow in the coupling can be eliminated by adding B Number lines to the f versus R diagram. Addition of these lines to the friction diagram makes possible a direct determination of the flow velocity V, which quantity is needed to determine the torque transmitted through the coupling and the power at each shaft.

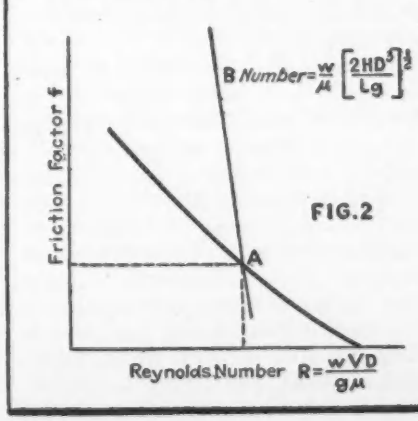


FIG. 2

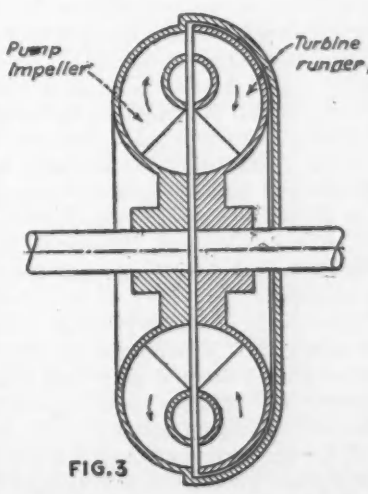


FIG. 3

Abrasive-Liquid Blast Produces Hone Finish

A. H. EPPLER
President, Vapor Blast Manufacturing Company

Fine abrasive suspended in a water-chemical emulsion and discharged by a spray nozzle at high velocity produces a finish comparable to that produced by honing. The process, originally intended for cleaning surfaces, is being used in numerous other applications because it is time saving and efficient in finishing surfaces.

A NEW surface cleaning and finishing method capable of producing a finish equivalent to that produced by honing, as measured by the Brush analyzer, is attracting the attention of design and production engineers. Dimensional tolerances of 0.0001 to 0.0002 in. are being maintained in some applications. The method consists of discharging an emulsion containing a finely powdered abrasive against metal surfaces at high velocity by means of compressed air. The process is known as Vapor Blasting or liquid honing. It gives metal surfaces a fine etched appearance.

The method was developed to improve upon the dry sand-blasting method of cleaning structural plate and to deter oxidation of the exposed virgin metal for a reasonable time after cleaning. One of the important advantages of suspending the abrasive in water is to prevent dust, thus making the process cleaner than sand blasting. The process has been found to have several other advantages which have extended its applications.

One of the principal advantages resulted early in the development of the process when it was discovered that particles of smaller mesh than silica could be used. After experiments with cereals and rice husks, a search for the finest abrasive obtainable led to a decomposed siliceous rock, which is available in powders and flours ranging in fineness from 80 to 2,500 mesh. Tolerance control can be maintained by varying the proportions and mesh of the abrasive in the emulsion.

The surface cleaning obtained with this material is much more thorough and of a different character than that obtained by ordinary sand blasting. Principal additional effect is removal of all foreign particles from the surface, which is an important factor in improving resistance to corrosion. This improvement is primarily dependent upon chemicals in the water but is also affected greatly by the abrasive used, as is shown by the fact that when silica is used instead of novaculite the corrosion resistance is lost.

Another important effect is the improvement of rubbing and bearing surfaces such as cylinder walls, guides, and cams. The coefficient of friction of a dry, liquid-honed surface is often less than a polished surface. Such surfaces also hold lubricant better than polished bearing surfaces, thus making them advantageous in bush bearings. This type of surface treatment has also been used to remove scuffing on the races of antifriction bearings to reclaim them.

One of the advantages of the method, as compared with other finishing methods by cutting, grinding, burnishing, or other similar methods whereby the surface is subjected to rubbing or tearing action, is that instead of covering over fine fatigue cracks, it exposes these flaws.

As a result of these and other effects the process has been used successfully on supercharger impellers for deburring and finishing, on articulated rods for aircraft engines to gain tensile strength, on guide vanes, connecting rod bolts, gear teeth, splines, gyroscope and other instrument parts. It has been used for finishing threaded parts, since it is possible to reach the bottoms of threads satisfactorily.

Deburring in the interior of intricate castings, where burrs cannot be removed economically any other way, has been an important application of the method. Applications of this type have resulted in great savings of time, manpower, and expense in production, which are important considerations in choosing the best design for a given part. In one plant the time for deburring was cut from 40 min. per piece by hand to 8 min. per piece by the Vapor Blast method.

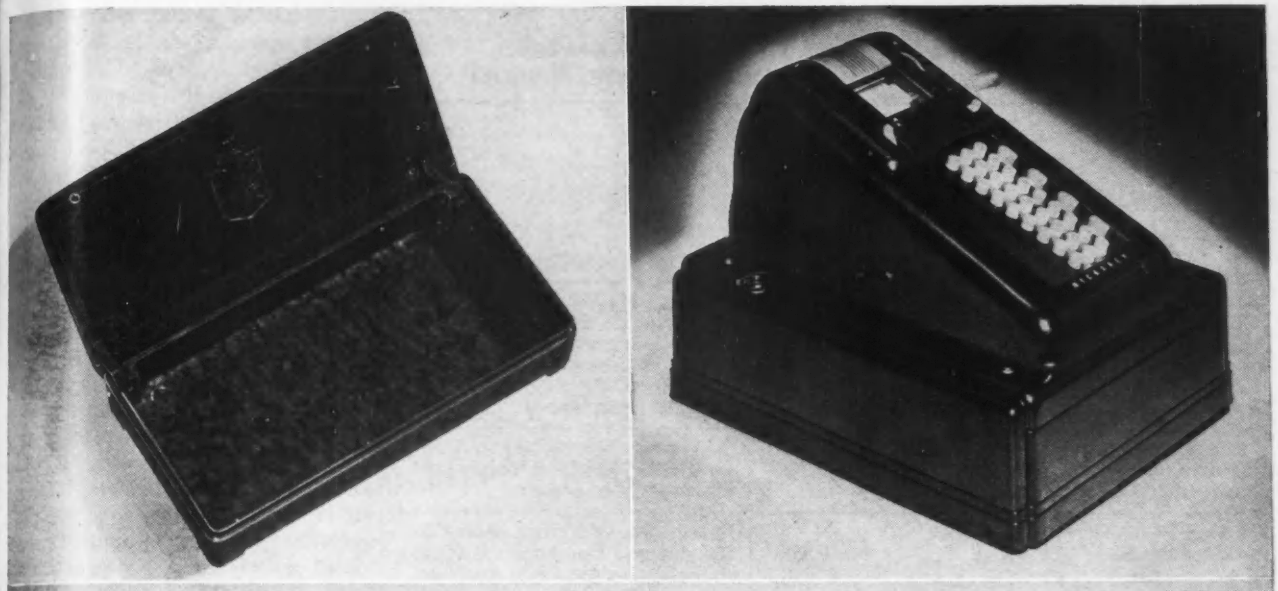
As a preparation for electroplating the process has been used in preference to pickling. It was found in chromium plating that flash spots, which commonly appeared on surfaces that had previously been pickled, were eliminated. The process is also being used in preparing surfaces for anodizing.

Gear teeth that are liquid honed after heat-treatment to remove scale and discoloration are not changed in their profile as is likely when hand polishing is employed. The same is true in removal of grinder marks.

In some applications it may be desirable to remove a measurable amount of metal from the surface. In such instances a two or three-step blasting schedule may be advisable: first using a coarse abrasive and then a finer abrasive for finishing. In designing these parts allowance should be made for the amount of stock to be removed, even though it be small. A simple method of shielding areas that are not to be blasted has also been worked out.

Liquid honing equipment consists of a cabinet inside which parts are blasted, a tank containing the abrasive and liquid mixture, a circulating pump, which serves also as an agitator to keep the tank content uniform, and an assortment of nozzles to suit various parts on which the process is used. The circulation system also permits metal oxides and foreign particles to settle out for periodic removal. In one production system liquid and abrasive are discarded after 40 hours service, make-up having been added at 8-hr. intervals during this time.

Cabinet design is more or less flexible so that the process can be fitted to individual requirements. It can also be adapted to automatic conveyor production lines as well as to tumbling barrels and other combinations of equipment. The process permits wide variation in abrasive, compressed air pressure, nozzle, size, abrasive mix and the water and chemical emulsion.



Bakelite Corp.

Molded phenolic is successfully used for containers and housings. Note uniform wall thickness, rounded corners and the molded-in lugs for hinging in the small container. Adding machine housing is lightweight, impact resistant and requires no finish except buffing.

Interpretation of Mechanical Data Necessary in Plastic Part Design

W. S. LARSON
Plastics Department, General Electric Company

Strength, shock resistance, elasticity and plasticity and hardness of representative plastic materials for parts under varying service conditions are discussed.

TEST DATA on mechanical properties of plastics parts are relatively difficult to convert into design data. One reason is that test data, embracing the numerous formulations and types associated with many plastics, cover such a wide range that the information is useless for design work. An analogy would be reporting the tensile strength of steel as 50,000 to 200,000 lb. per sq. in. without differentiating between the types of steel. Another reason is that most plastics are more sensitive to operating conditions than metals. Test data are necessarily gathered at some arbitrary operating condition, and comparatively slight divergences from these conditions can greatly change the observed value for that material.

This discussion of mechanical properties is broken down into strength, shock resistance, elasticity and plasticity, and hardness. However, this listing is for convenience only. It is impossible to provide the factual data necessary to cover a wide range of materials and operating conditions so that it is necessary that the designer, by knowing the material he is working with and its characteristics, exercise good judgment.

Influencing Characteristics

Thermosetting resins polymerize into a rigid three-dimensional framework. Intermolecular bonds are cross linked to the extent that an entire molded part might conceivably be one large macromolecule. These resins are strong, rigid, brittle and have a high heat-distortion point. Any stress tending to deform the material is absorbed by the intermolecular bonds being strained within the fields of their force. Fillers are usually added to lower cost and shrinkage and to improve certain properties. Thus, materials such as the phenolics, ureas and melamines are made up of filler particles of woodflour, cotton flock or asbestos or rag fibers firmly bound into a rigid resin framework. The outstanding mechanical function of fillers is to improve the shock resistance of the resin.

Most common thermoplastic resins polymerize end to end into linear molecules which, in turn, are held together mainly by a purely mechanical interweaving. If the resin is plasticized the interwoven molecules are freer to move over each other and the material is more easily deformed. When these molecules

Table I—Examples of Test Values and Allowances For Service Conditions for Representative Plastic Materials

Material	Standard A.S.T.M. Tests			Remarks
	Tensile Strength, lb. per sq. in.	Compressive Strength, lb. per sq. in.	Flexural Strength, lb. per sq. in.	
Molded Phenolic, woodflour filled—general purpose	8,000	20,000 to 35,000	10,000	Tensile strength decreases ⅓ at 160 deg. C. Flexural strength begins marked decrease at 150 deg. C. long time. Compressive strength shows wide range for different formulations.
Molded Phenolic, rag-fiber filled—impact material	6,200	20,000 to 32,000	10,000	Same behavior as woodflour filled. Long-time tensile strengths approximately 40 percent.
Molded Phenolic, asbestos filled—heat resistant	6,000		9,000	Retains strength when heated to 200 deg. C. long time. Tensile strength drops slightly when tested at 240 deg. C.
Laminated Phenolic—NEMA "L," cotton-fabric filler	16,000	25,000		Long-time tensile strength 70 percent.
Molded Urea, alpha-cellulose filled	6,000	24,000	11,000	Sensitive to long-time heating.
Cellulose-Acetate Sheet, Air Corps Specification 12025-B	5,800			Tensile strength 10,000 lb. per sq. in. at −20 deg. C., 16,000 lb. per sq. in. at −40 deg. C. Practical compressive strength limit is 4,000 lb. per sq. in.
Cast Polymethyl Methacrylate	8,000	12,000	15,000	Long-time tensile strength 50 percent. Recommended long-time flexural strength 1,000 lb. per sq. in. Compressive strength drops to 4,000 lb. per sq. in. at 60 deg. C.
Injection-Molded Polystyrene	7,000	15,000	17,000	No strength over 80 deg. C. for practical use.

are crimped or spiralled and are sufficiently plasticized they have the faculty of stretching as much as 500 percent and returning to their original shape. By definition, all thermoplastic resins are sensitive to heat and practically all are mechanically unreliable at 80 deg. C. By mechanically orienting the linear molecules of a thermoplastic resin, filaments and sheets can be made which have extremely good tensile strengths.

Evaluating Strength

The internal structure of a plastic acts something like a dashpot with a rubber band tied to the plunger. If the rubber is pulled quickly it will stretch and restore itself when released, pulled slowly, the dashpot plunger will move a little and not tend to restore itself. Thus, plastics respond to the rate of loading, humidity and temperature, which affect its dashpot action.

Values representing the strength of plastic material must be closely considered in the light of the conditions under which tests were made.

Ordinary materials show these same characteristics of elasticity and plasticity, but to a lesser degree. Although standard tests specify the method of preparing the sample, the temperature, humidity and testing speed, it is the designer who translates properly from test to design values. Examples of test values are presented in Table I along with some of the allowances for service conditions.

Increases in temperature from −80 to 160 deg. C. will lower the tensile and flexural strengths of phenolics from 10 to 40 percent of their test values. Most thermoplastics have strengths twice as high as test values at −40 deg. C. and strengths decreasing to 0 at temperatures near 100 deg. C. Test values are ordinarily taken at 25 deg. C. Specimens subjected to high relative humidities or complete immersion in water may show a slight decrease in strength, but usually not over 10 percent.

Plastic parts in service will be stressed at different rates and, in most applications, will be loaded for long periods of time. An example of the effect of the rate of stressing is the test of cast polymethyl methacrylate. The tensile strength will drop 20 percent if the load is applied at the rate of 0.05 in. per min. as compared to a rate of 0.3 in. per min. It will drop 50 percent if applied at an infinitely slow rate, or the equivalent of long-time loading. Molded phenolics will average only 40 percent of their short-time test value when they are loaded for 1,000 hr. Phenolic laminates can be loaded at 70 percent of their short-time value. Loads applied rapidly exhibit entirely different characteristics, which are discussed under shock resistance.

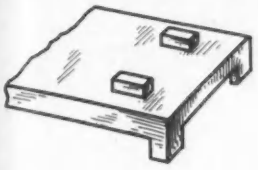
Thermosetting materials that are held at elevated temperatures for long periods of time lose their mechanical strengths because the cellulosic filler or the resin itself decomposes. Marked decreases in strength are evident after 200 hr. at 140 deg. C. for fabric-filled phenolics. Mineral-filled phenolics are comparatively safe up to 220 deg. C. Thermoplastics may show an equivalent long-time loss at much lower temperatures because of loss of volatile plasticizers, as evidenced by shrinking and cracking.

Filaments of thermoplastics, notably Nylon and vinylidene chloride, can be drawn to one-half of their diameter and thus elongated 400 percent by a mechanical stretching. This orients the molecules lengthwise of the filament and may increase the tensile strength from 8,000 lb. per sq. in. to as high as 60,000 lb. per sq. inch.

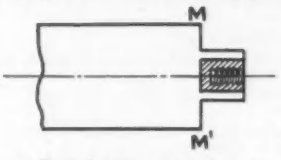
Shock Resistance

Most plastic parts are not designed for service on the basis of their tensile and flexural strengths, but on a basis of shock resistance. When a plastic is struck sharply, strains in tension and compression are set up much as though the part was being stressed in flexure. The outstanding difference is that the load is applied so suddenly that it does not have a chance to equalize itself throughout the cross-sectional area. Instead, the stress is concentrated at some surface irregularity and failure occurs progressively through the part. So many considerations enter into impact testing that it is impossible to ascertain more than very rough design values. They will be based on test values, such as those listed in Table II, and modified by allowances for variations in tempera-

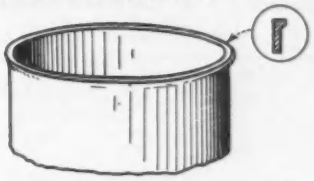
MECHANICAL DESIGN DETAILS FOR PLASTIC PARTS



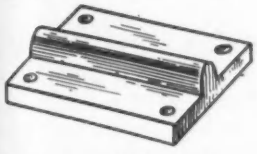
Projections and sharp corners without fillets have poor shock resistance



End of insert and sharp corners form a weak section across M-M'

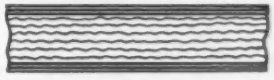


Small beads are especially effective in improving shock resistance of thin edges

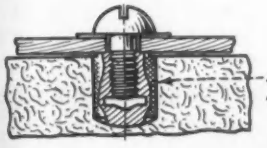


Judicious use of molded ribs improves design and strength of plastic parts

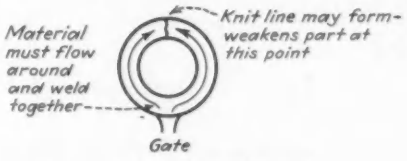
Thin sections and covers are stronger when domed



Laminated sandwiches of low-density core and high-strength skins provide light and strong structural material

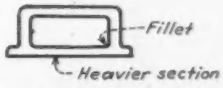


Brass inserts, sizes 10 to 3/8 in. and similar to that shown, will withstand from 2,000 to 4,000 lb. ultimate tensile pull. Molded-in inserts should project above surface so that the fastening bears on metal and does not tend to pull the insert out of the plastic and cause cold flow



Proper gating of transfer and injection molded parts, proper density and cure of molded parts and elimination of strains and cracks are necessary to get best mechanical properties

Strains caused by shrinkage are set up by extreme and abrupt changes in cross section. Thin ends will chip.



Better design

Table II—Typical Values for Izod Impact Strength of Plastic Materials

Material	Ft. lb. per inch of notch	Remarks
Molded Phenolic, woodflour filled—general purpose	0.3	Loses strength under continuous heat over 150 deg. C. For repeated impact use 25 percent of this value. No loss at low temperature.
Molded Phenolic, rag-fiber filled—impact material	1.0 to 3.0 depending on length of fiber	Do not use under continuous heat over 150 deg C. Cord-filled molding compounds have higher impact strength.
Molded Phenolic, asbestos filled—heat resistant	0.3	Can be used continuously at 200 deg. C. No loss in strength at low temperature.
Laminated Phenolic—NEMA "C", cotton-fabric filled	3.5 typical 15.0 maximum	For repeated impact use 50 percent of these values.
Molded Urea, alpha-cellulose filled	0.3	Do not use under continuous heating over 80 deg. C.
Injection-Molded Cellulose Acetate Butyrate, medium-soft flow	2.2	Higher for softer flows. Marked decrease at lower temperature. For repeated impact use 5 percent of this value.
Cast Polymethyl Methacrylate	0.3	No loss in strength at −80 deg. C. For repeated impact use 40 percent of this value.
Injection-Molded Polystyrene	0.3	No loss in strength at −80 deg. C.

ture, humidity, stress concentration and fatigue.

The accepted standard for impact tests at present is the Izod test, in which a specimen is broken. Test values are then reported as the number of foot pounds required to break the specimen, or as the number of foot pounds per inch of notch. If the specimens are each ½ in. wide, the latter value will be twice that of the former. Haphazard listing of this value often fails to state which value is being expressed. By choosing the best material indicated by this test and designing for shock resistance, the designer may feel that he has done the best job possible without making an extended study of the subject.

Most pertinent design feature is the elimination of sharp corners and other constructions which concentrate the stress. Shock resistance is not synonymous with impact strength; shock may exert pure tensile, compressive and shear stresses and must be given consideration in many cases. A typical instance is a plastic component for ammunition which must withstand the terrific setback forces of the firing of the projectile. Another instance is the woven Nylon tow ropes being used for gliders which are subject to a tensile shock.

Impact-grade molded phenolics are expected to maintain approximately

Table III—Typical Resistance Values For Various Plastic Materials.

Highest values represent hardest materials

Material	Resistance to Indentation		Scratch Resistance		Abrasion or Mar Resistance
	Brinell	Rockwell M	Mohs	Beirbaum	Tentative A.S.T.M. Glossmeter, arbitrary rating
Molded Phenolic, woodflour filled—general purpose	30–45 [1]	100–125			
Molded Phenolic, mineral filled—heat resistant		85–120			
Molded Urea, alpha-cellulose filled		110–130			
Laminated Urea-Melamine			3–3.5	31	77
Cast Polymethyl Methacrylate	18–20 [2]	85	2–3	18	61
Injection-Molded Polystyrene	20–30 [1]	75–90		16	39
Vinyl Copolymer, rigid sheet	12–15 [2]			12	65

[1] 2.5-mm. ball, 25-k. load. [2] 10-mm. ball, 500-kg. load.

these ratios of impact strengths under the following conditions:

Heated 24 hr. at 100 deg. C.	1.00
96 hr. at 25 deg. C. and 70 percent relative humidity	1.10
48 hr. immersion at 50 deg. C.	1.25
5 hr. at —55 deg. C.	0.75

Laminated plastics have marked unidirectional properties and specimens tested edgewise will have only 30 percent of the impact strength of specimens tested flatwise. Scratches 0.025 in. or more deep greatly reduce the impact strength of polymethyl methacrylate.

Most plastics show poor resistance to repeated blows. Design values should range from ½ to 1/20 of the single-blow value as listed in Table II. Proof tests are frequently used in order to check designs for shock resistance. One method of doing this is to drop the part in a prescribed manner from successively higher levels until it fractures. An arbitrary minimum is set, based on requirements or previous performance.

Elasticity and Plasticity

A study of the elastic and plastic make-up of plastics gives a further insight to their mechanical properties. As a stress—compressive load, for example—is placed on a plastic sample, the inner molecular structure will tend to deform to accommodate the strain. As heat passes into the part from the surrounding medium the inner deformation of molecular forces tends to relax. This process then repeats itself and a noticeable permanent set becomes evident. This accounts for the phenomena of creep and cold flow. It is obvious that the higher the stress and the higher the temperature, the more extreme will be the cold flow.

Cold flow is an important consideration where tolerances are to be held on parts under stress. The most obvious failures are those in which a fastening, such as a bolt or clamp, fails as the plastic material flows from under it. This is also an important consideration in determining bearing loads. Phenolic materials show a correlation between cold flow and impact strength; those with high impact strength also have a high cold flow. Rag-fiber and woodflour filled phenolics should not be stressed above 1,000 lb. per sq. in., or mineral filled above 2,000 lb. per sq. in. if the tolerances or performance are critical. At temperatures of 100 deg. C., loadings should be ¼ of these values. Thermoplastics are particularly susceptible to cold flow, especially at slightly elevated temperatures. Cast methacrylate sheet shows a fair rigidity up to 3,000 lb. per sq. in. at 25 deg. C. Where the best performance is required the harder or unplasticized formulations should be used.

Hardness

Hardness is generally conceded to be the resistance offered by a material to surface indentation or abrasion. No single test, however, will show that a material resists indentation, abrasion and scratching and at the same time express its rebound efficiency. It is well to eliminate the term "hardness" and define each of these properties independently. Each fills a unique function in evaluating the usefulness of a material. Table III lists these values for comparison. There is no relation between the values of different test methods. Probably the most useful value is that for abrasion or mar resistance. It determines the practicability of a material for specific uses.

The most important considerations in designing for mechanical service are:

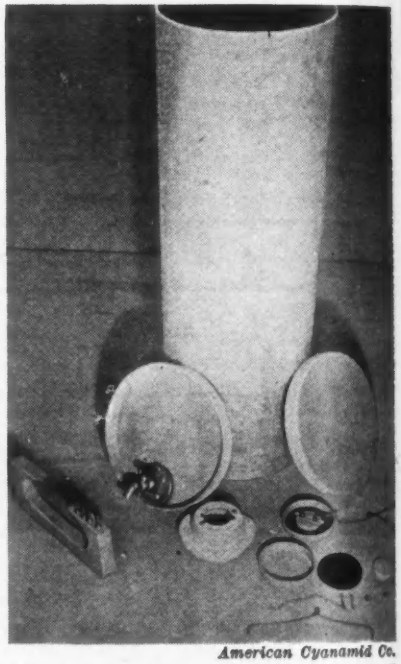
1. Test values are taken at an arbitrary condition and wide allowances must be made for service values.

2. Test values for a given plastic will usually duplicate within 10 percent.

3. High temperature and shock loads are common causes of mechanical failure.

4. Mechancial properties can be improved by careful design of wall thicknesses, radii and fillets.

5. Material formulations can be specified over a wide range, but improvement of hardness and tensile strength usually means a loss of shock resistance and elongation.



American Cyanamid Co.

Low pressure laminating is used to advantage in the design of this water tank shown disassembled. End covers supply reinforcement to the tank structure.

A.G.M.A. Tentative Standard System Fine-Pitch Straight Bevel Gears

This Tentative Standard was adopted by the American Gear Manufacturers Association at their annual meeting, May 22–24, 1944.

Section I—Introduction

This standard for fine-pitch straight bevel gears follows the same general principles as the fine-pitch spur gear system. The tooth proportions are similar to those in the A.G.M.A. Standard Tooth Forms—"Straight and Spiral Bevel Gears" No. 202.01, but with the following modifications:

1. Clearances are increased.
2. Intervals of addendum variations are condensed into a simplified table.
3. Tooth thicknesses correspond to those generated by a crown gear in which the tooth thickness and space width are equal.

The face width is limited to one quarter of the cone distance.

Section II—Scope and Limitations

1. While theoretically the fineness of pitch in straight bevel gears is limited only by the point width of the tool, actually it is not considered practical to generate these gears in pitches finer than 120 diametral pitch. In practice most of the fine-pitch straight bevel gears lie between 30 diametral pitch and 64 diametral pitch and this range is to be preferred wherever possible.

2. This standard covers both right-angle and angular bevel gears.

3. Gears with 13 teeth are the smallest miters in this system. There is no reduction of contact ratio due to undercut. The theoretical contact ratio is not less than 1.5.

4. The smallest pinion has been established as having 12 teeth and may be meshed with a gear having 13 or more teeth.

Section III—General Specifications

Table I—General Dimensions

Linear dimensions are in inches

Scope of system	30 diametral pitch to 120 diametral pitch
Pressure angle	$\phi = 20$ degrees
Smallest pinion	12 teeth
Smallest 90 deg. miter	13 teeth
Where P = diametral pitch	
Working depth	$h_k = 2.000/P$
Whole depth	$h_t = (2.200/P) + 0.002$
Maximum face width	F = cone distance $(A_o)/4$

(continued on next page)

Section IV—Nomenclature and Symbols

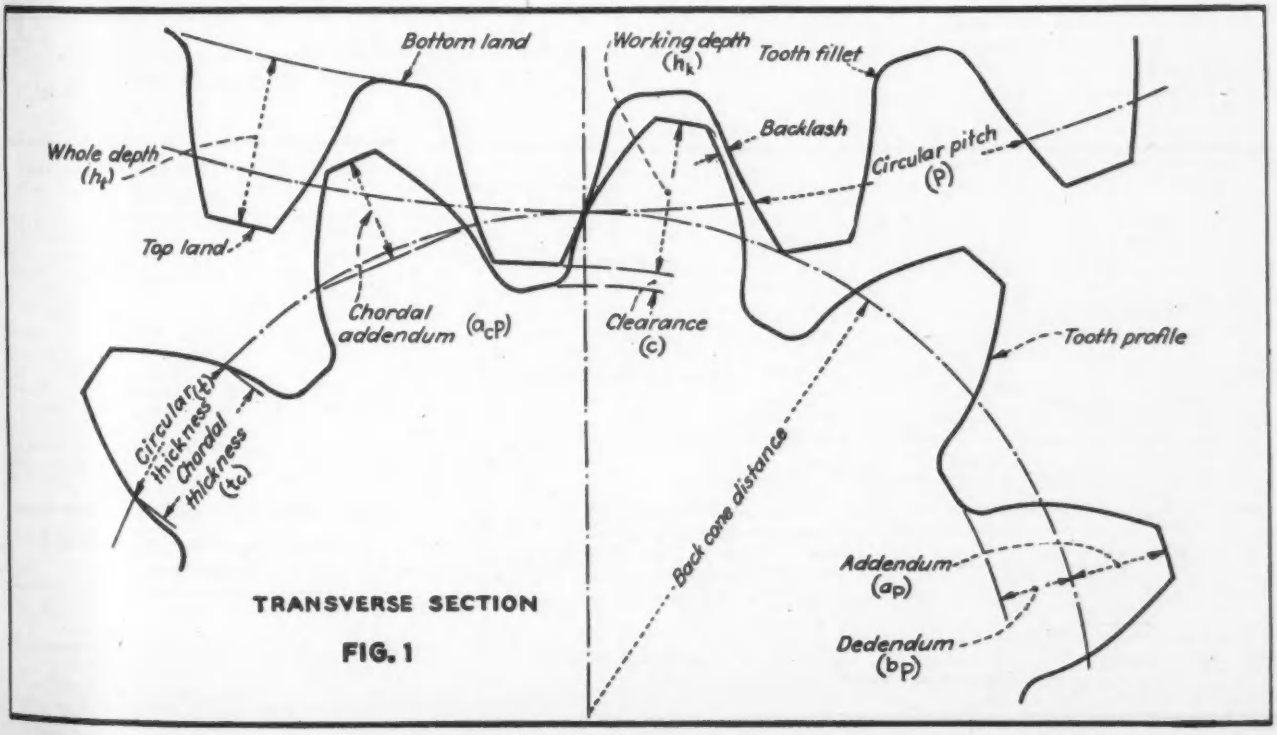


Fig. 1—Nomenclature and Symbols

Section V—90-Degree Axis Bevel Gears

1. TOOTH PROPORTIONS (RIGHT ANGLE BEVEL GEARS)

Table II—Tooth Proportions for 1 Diametral Pitch

(Axes at 90 Deg.)

$$\text{Ratio} = \frac{\text{Number of gear teeth}}{\text{Number of pinion teeth}}$$

Ratio	Pinion Addendum	Gear Addendum	Pinion Circular Thickness	Gear Circular Thickness
1.00	1.000	1.100	1.5708	1.5708
1.01 to 1.14	1.100	0.900	1.6436	1.4980
1.15 to 1.33	1.200	0.800	1.7164	1.4252
1.34 to 1.70	1.300	0.700	1.7892	1.3524
1.71 to 2.78	1.400	0.600	1.8620	1.2796
2.79 and higher	1.450	0.550	1.8984	1.2432

Dimensions are in inches.
Where pinions and gears are to be hardened, consideration of top lands may require modification of addendum proportions.

2. GEAR BLANK DIMENSIONS AND TOLERANCES

In dimensioning bevel gear blanks, it is necessary to specify properly the items important to the functioning of the teeth. These are:

1. Outside diameters
2. Crown to back (or mounting surface)
3. Face angle
4. Mounting distance

Face width and back angle are usually of secondary importance.

Recommended tolerances are listed for outside diameter, crown to back, face angle and back angle. The first three directly affect the addendum height and therefore the contact ratio.

(a) Outside Diameter and Crown to Back

These tolerances apply to the theoretical crown point at which the face cone and back cone elements intersect. The outside diameter cannot, therefore, be measured directly if the blank has been rounded or flatted. See Fig. 3.

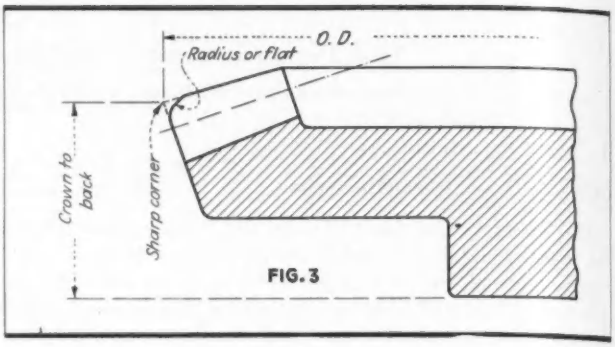


Fig. 3—Outside diameter and crown to back dimensions apply to the theoretical crown point.

(b) Face Angle

The face angle is made so that the face cone element is parallel to the root cone element of the mate. This produces constant clearance along the tooth. See Fig. 4.

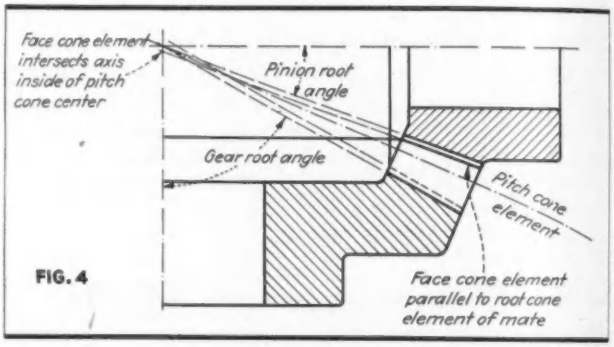

Face cone element intersects axis inside of pitch cone center
Pinion root angle
Gear root angle
Pitch cone element
FIG. 4
Face cone element parallel to root cone element of mate

Fig. 4—Each face cone element is made parallel to the respective root cone element of its mating gear.

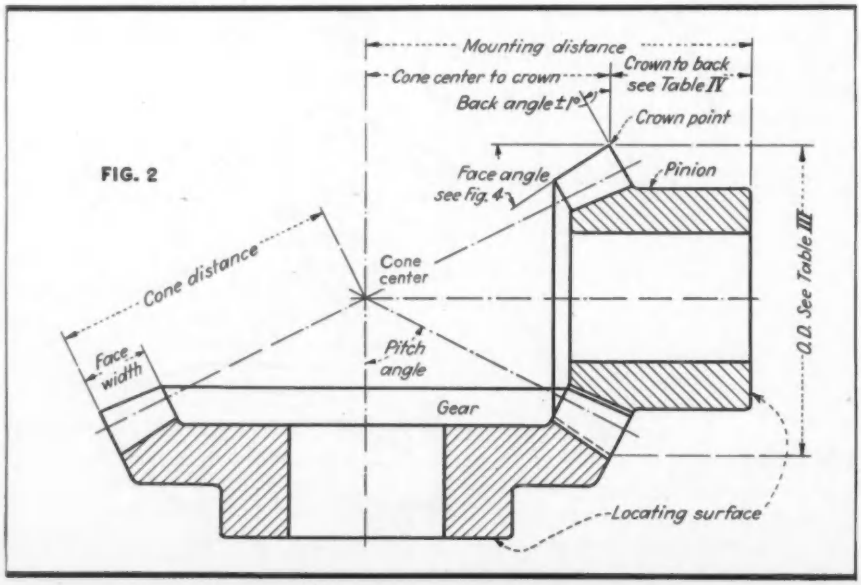


Fig. 2—Important bevel gear blank dimensions.

Table III—Outside Diameter Tolerance

Diametral Pitch	Tolerance Inches
30 to 40	+0.000 −0.004
41 to 56	+0.000 −0.003
57 to 94	+0.000 −0.002
95 to 120	+0.000 −0.001

Table IV—Crown to Back Tolerance

Diametral Pitch	Tolerance Inches
30 to 47	+0.000 −0.002
48 to 120	+0.000 −0.001

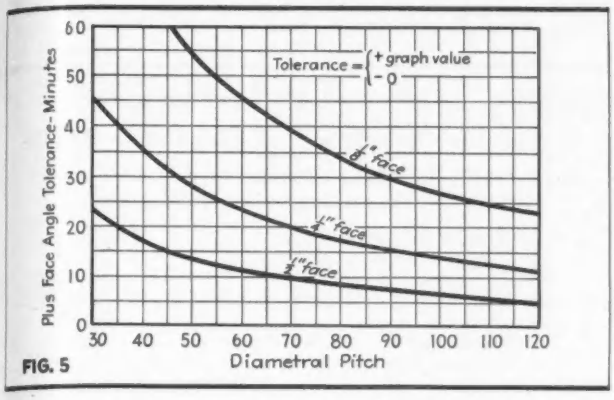


FIG. 5

Fig. 5—Graph of face angle tolerance. Values for other face widths are obtained by interpolation.

The face angle tolerance is taken from the graph, Fig. 5. Values for other face widths are obtained by interpolation. Specify tolerance to nearest 10 minutes. In establishing these tolerances consideration was given to the increase in clearance and the corresponding tooth height.

(c) Mounting Distance

1. Bevel gears are cut and inspected to the mounting distance. In order to obtain the running characteristics machined into the gears, they should be assembled to the same dimension. Variations affect tooth bearing and backlash. The results of errors in mountings became more pronounced in low ratio combinations. Allowable tolerance for mounting distance will, therefore, depend on the design and accuracy required of the application.

2. Exact position for assembling bevel gears should be determined by the mounting distance. It is not good practice to depend on setting the back cones flush.

(d) Back Angle

A tolerance of ±1 degree will generally be satisfactory.

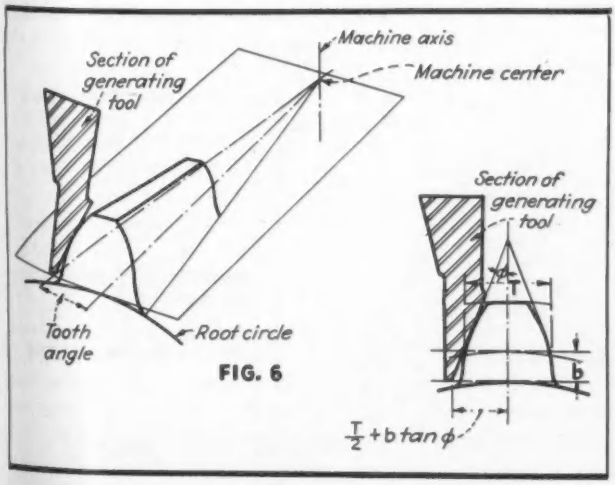


FIG. 6

Fig. 6—The tooth angle is the angle at the machine center between the line to the center of the tooth and the line to the point of the tool.

Table V—Fine Pitch Straight Bevel Gear Dimensions

Ref. to Figs. 1, 2, 4 and 6. Linear dimensions are in inches. Axes at right angles.

1. Number of Pinion Teeth..... n
2. Number of Gear Teeth..... N
3. Diametral Pitch..... P
4. Face Width..... F
5. Working Depth..... $h_k = \frac{2.000}{P}$
6. Whole Depth..... $h_t = \frac{2.200}{P} + 0.002$
7. Pressure Angle..... $\phi = 20$ deg.

	Pinion	Gear
8 Pitch Diameter	$d = \frac{n}{P}$	$D = \frac{N}{P}$
9 Pitch Angle	$\gamma = 90° - \Gamma$	$\Gamma = \tan^{-1}\frac{N}{n}$
10 Cone Distance	$A_o = \frac{D}{2 \sin \Gamma}$	
11 Addendum	$a_P = \frac{\text{Table II}}{P}$	$a_G = \frac{\text{Table II}}{P}$
12 Dedendum	$b_P = h_t - a_P$	$b_G = h_t - a_G$
13 Dedendum Angle	$\delta_P = \tan^{-1}\frac{b_P}{A_o}$	$\delta_G = \tan^{-1}\frac{b_G}{A_o}$
14 Blank Face Angle	$\gamma_o = \gamma + \delta_G$	$\Gamma_o = \Gamma + \delta_P$
15 Root Angle	$\gamma_R = \gamma - \delta_P$	$\Gamma_R = \Gamma - \Gamma_G$
16 Outside Diameter*	$d_o = d + 2\,a_P \cos\gamma$	$D_o = D + 2\,a_G \cos\Gamma$
17 Cone Center to Crown*	$\frac{D}{2} - a_P \sin\gamma$	$\frac{d}{2} - a_G \sin\Gamma$
18 Circular Thickness	$t = \frac{\text{Table II}}{P}$	$T = \frac{\text{Table II}}{P}$
19 Chordal Thickness*†	$t_c = t - \frac{t^3}{6d^2}$	$T_c = T - \frac{T^3}{6D^2}$
20 Chordal Addendum*	$a_{cP} = a_P + \frac{t^2 \cos\gamma}{4d}$	$a_{cG} = a_G + \frac{T^2 \cos\Gamma}{4D}$
21 Tooth Angle (see Fig. 6)	$\frac{3438}{A_o}\left(\frac{t}{2} + b_P \tan\phi\right)$ minutes	
22 Limit (max) Point Width	$\frac{A_o - F}{A_o}\left(t - 2b_G \tan\phi\right)$	

* Calculations which affect angular values are carried to four or five decimal places to obtain the required accuracy. Final dimensions on drawings and specifications are given in thousandths.

† No backlash is indicated since most fine pitch gears operate with little backlash. If backlash is desired, subtract one half of the desired amount from each chordal thickness.

(continued on next page)

Section VI—Angular Bevel Gears

Bevel gears with shaft angle different from 90 degrees.

1. PITCH ANGLES

Tooth proportions are obtained in a different manner than for right angle bevel gears.

Given the shaft angle and the number of teeth, the pitch angles are found from the following general formulas:

$$\tan \gamma = \frac{\sin \Sigma}{\frac{N}{n} + \cos \Sigma}$$

$$\Gamma = \Sigma - \gamma$$

where Σ = shaft angle
n = number of teeth in angular pinion
N = number of teeth in angular gear
γ = pitch angle of angular pinion
Γ = pitch angle of angular gear

Note that cos Σ is negative when Σ is more than 90 deg. If the pitch angle Γ is larger than 90 deg., the gear is internal and special methods of cutting are required.

2. EQUIVALENT NUMBERS OF TEETH

In order to use Table II for tooth proportions, it is necessary to calculate the ratio of equivalent 90 deg. bevel gears. The equivalent 90 deg. bevels may be defined as the right angle bevel gears whose respective back cone distances are equal to those of the angular bevels. See Fig. 7.

The formula for the ratio m_{90} of the equivalent 90 deg. bevel gears is:

$$m_{90} = \sqrt{\frac{N \cos \gamma}{n \cos \Gamma}}$$

The dimension, cone center to crown, (Item 17, Table V) must be calculated as follows for angular bevel gears:

Cone Center to Crown

Pinion $= A_o \cos \gamma - a_P \sin \gamma$
Gear $= A_o \cos \Gamma - a_G \sin \Gamma$

3. EXAMPLE

Given: Shaft angle 120 deg., 20 tooth pinion, 30 tooth gear, 30 diametral pitch, face width ⅛ inch.

$\sin 120° = 0.866025$
$\cos 120° = -0.500000$

$$\tan \gamma = \frac{0.866025}{\frac{30}{20} + (-0.500000)} = 0.866025$$

$\gamma = 40.893° = 40° 54'$
$\sin \gamma = 0.65465$
$\cos \gamma = 0.75593$
$\Gamma = 120° - 40° 54' = 79° 6'$
$\sin \Gamma = 0.98198$
$\cos \Gamma = 0.18898$

$$m_{90} = \sqrt{\frac{30 \times 0.75593}{20 \times 0.18898}} = 2.4$$

Table VI—Tabulation of Calculated Results

Blank Dimensions (Refer to Table V)
$\Sigma = 120°$ Linear dimensions are in inches.

1. Number of Pinion Teeth	20
2. Number of Gear Teeth	30
3. Diametral Pitch	30
4. Face Width	1/8
5. Working Depth	0.0666
6. Whole Depth	0.0753
7. Pressure Angle	20 deg.

	Pinion	Gear
8 Pitch Diameter	0.6667	1.0000
9 Pitch Angle (Special for Angulars)	40°54′	79°6′
10 Cone Distance	0.5092	
11 Addendum ($m_{90} = 2.4$)	0.0466	0.0200
12 Dedendum	0.0287	0.0553
13 Dedendum Angle	3°14′	6°12′
14 Face Angle	47°6′	82°20′
15 Root Angle	37°40′	72°54′
16 Outside Diameter	0.737	1.007
17 Cone Center to Crown	0.354	0.077
18 Circular Thickness	0.0621	0.0426
19 Chordal Thickness	0.062	0.042
20 Chordal Addendum	0.048	0.020
21 Tooth Angle	4°40′	
22 Limit Point Width	0.016	

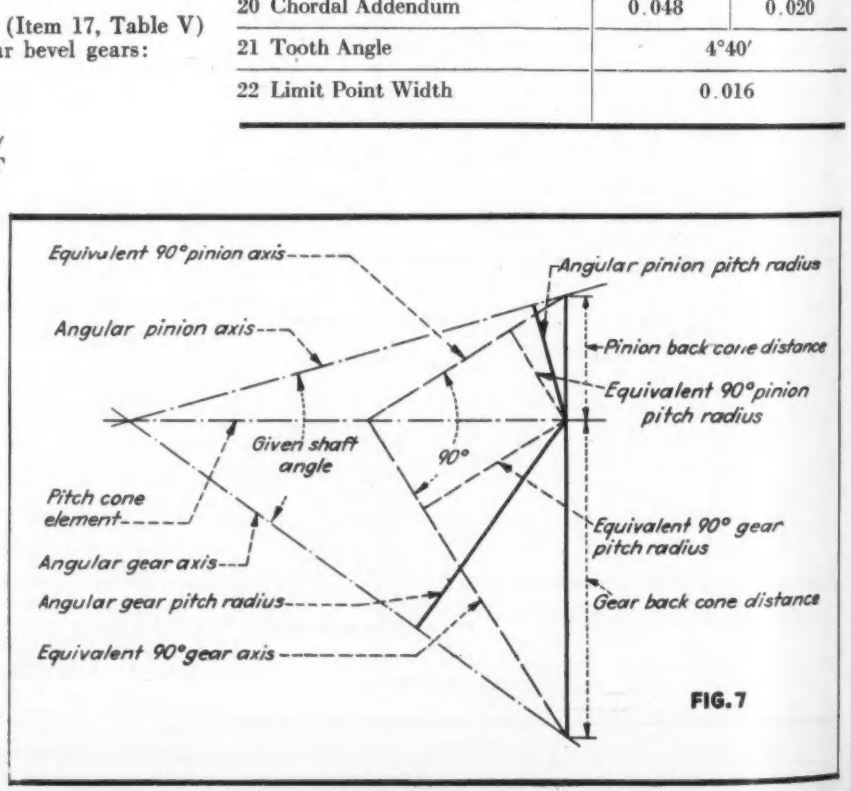


Fig. 7—Equivalent 90-deg. bevel gear.

Tolerances and Allowances For Interchangeable Assembly

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Interchangeable assembly, as affected by tolerances on the dimensions that locate the mating elements, is considered in relation to the tolerances of the male and female dimensions. Algebraic relationships between least play in attaching elements and tolerances on dimensions locating the attaching elements are derived. Included are tables of tolerances that can be used for commonest bolt sizes when attaching elements contain two or more holes.

THE fine points of interchangeability in assembly often do not come to the attention of design engineers when nearly all component parts of an assembly are fabricated in one plant. This may be because difficulties resulting from producing tolerances on the low side on one part and on the high side on a part to be joined to it are often quickly ironed out in the shop without necessity for revision of drawings. When various parts of an assembly are fabricated in widely separated plants, the necessity for most careful selection of tolerances and clearances becomes an increasingly important design problem.

As an example of what may happen without proper dimensional tolerances, two components of a machine are joined by two bolts through holes in each of two plates. Maximum diameter of bolts is 0.249 in. and minimum diameter of holes is 0.250 in. The two holes are dimensioned 2 in. center to center. Since it is impossible to hold the dimension exactly, limits are specified; for example, an upper limit of 2.005 and a lower limit of 1.995 in. If one plate is made to the upper limit and one to the lower, and if the holes are at the minimum allowable, 0.250 in., a mismatch like that in Fig. 1 will have an opening width, $R = 0.245$ in. Obviously, the bolts will not go through the opening and assembly is impossible without additional operations.

There are various operations which permit assembly of mismatched parts: (1) Run a drill or reamer through the misaligned bolt or rivet holes to enlarge them; (2) discard one plate and find another with an oversize hole or ream the hole in one of the plates, if tolerances on hole diameter permit; (3) substitute small diameter bolts, 0.245 in. in diameter in the example cited, if stress limits permit; and (4) lay aside the 1.995-in. plate and find another with holes 2.005 in. apart. However, none of these is permissible if interchangeability is required. To assure full interchangeability, it is necessary to increase tolerances on the hold diameters or lower tolerances on the center distance.

The easiest way to redesign is to increase the size of the holes 0.010 in. or more, which insures that the bolts can be installed. But the amount of clearance between bolt and holes is often critical to the strength of the attachment. Therefore, it is desirable to increase hole size just enough to allow assembly.

Definition of Terms

In deriving precise relationships between tolerances on locating dimensions and clearances in the bolt-hole fit, the terms limits, tolerance, clearance and allowance are defined:

1. Limits of a dimension are the least and greatest values that are acceptable for that dimension.

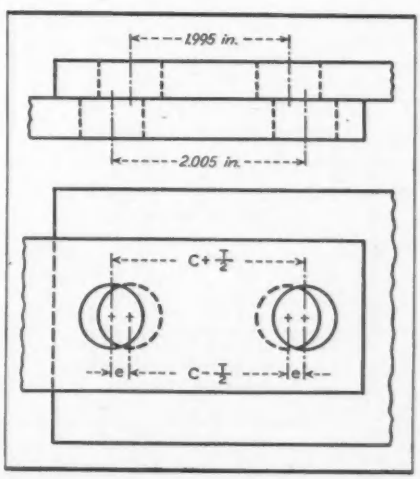


Fig. 1—Pattern of two holes in each of two mating parts with maximum mismatch. This occurs when hole center distance is at the upper limit on one plate and at the lower limit on the other.

2. Tolerance on a dimension is the difference between the upper limit and the lower limit.

3. Clearance in the fit of two mating parts is the difference between the inside dimension of the outside part and the outside dimension of the inside part.

4. Allowance refers to the least clearance resulting from any combination of limits on the dimensions of each of two mating parts.

Definitions for clearance and allowance are generalized from definitions for cylindrical fits. All four definitions agree with American Standards Association usage in B4a-1925 and with Sheffield Corporation "Dimensional Control." The relationship problem discussed here differs from that presented by J. Gaillard of the A.S.A. in "Tolerances for Cylindrical Fits." Material presented in engineering handbooks is of the same nature as the two mentioned A.S.A. references, the tolerances referring to the mating dimensions of male and female fits and affecting only the class of fit. In this article the tolerances on the dimensions that locate the mating elements are considered in relation to the tolerances of the male and female dimensions. Interchangeable assembly is affected as well as the class of fit. Formulas have been developed for common problems.

Two general kinds of attachment are: (1) Attachment by two or more bolts through matching hole patterns. (2) Attachment by a combination of two or more clevis-type joints.

Hole Patterns

When all holes in both parts and all bolts are the same nominal size, the following notation is used:

D_H = least diameter of hole; also nominal diameter of hole with positive unilateral tolerance.
D_B = greatest diameter of bolt; also nominal diameter of bolt with nega-

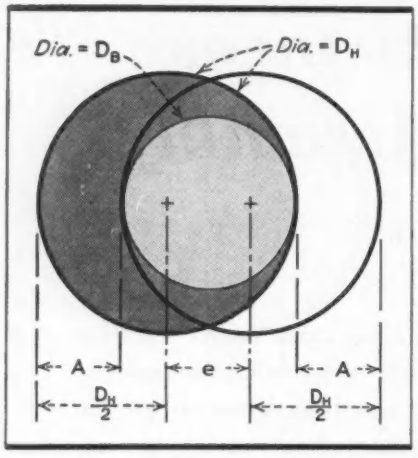


Fig. 2—Two superimposed, noncoincident holes, one in each of two parts, with dimensions for a fundamental formula.

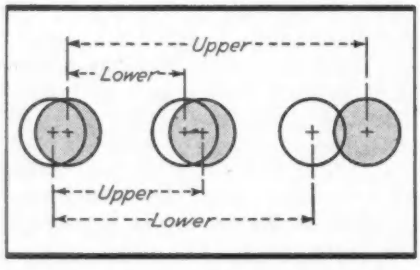


Fig. 3—Pattern of three holes in each of two mating parts, one part being shifted so as to divide the mismatch equally between two of the three holes.

tive unilateral tolerance.

A = allowance, the minimum clearance between bolt and hole.

C = distance between centers of holes. In patterns of more than two holes, subscripts are used.

T = tolerance on C, will be bilateral $\left(\pm\frac{T}{2}\right)$; subscripts correspond to C; when different tolerances exist on corresponding dimension on different parts, they are distinguished by such symbols as T', T''.

By definition for allowance:

$$A = D_H - D_B \qquad (1)$$

Two noncoincident holes, one in each of two parts, are superimposed in Fig. 2. A fundamental formula, which will be used in later derivations, is immediately obvious:

$$\tfrac{1}{2}D_H + e + \tfrac{1}{2}D_H = A + D_B + A$$

or
$$e = 2A - (D_H - D_B)$$

Using Equation (1)
$$e = A \qquad (2)$$

EXAMPLE 1: Pattern of Two Holes

Fig. 1 shows holes in two parts, two holes in each, when the distance between centers in one part is at the upper limit and in the other part at the lower limit. It is obvious that

$$C + \frac{T}{2} = e + C - \frac{T}{2} + e$$

$$2e = T$$

or using Equation (2),
$$T = 2A \qquad (3)$$

EXAMPLE 2: Pattern of More than Two Holes

Equal division of mismatch does not solve the problem. If one part is shifted so as to divide the mismatch equally for one pair of holes whose center distance is on the upper limit, mismatch of another pair whose center distance is on the lower limit will be aggravated. Fig. 3 illustrates this condition. To avoid this, the hole in the pattern, from which the others are dimensioned, must have its center coincident with its matching hole. Then the center distance between this hole and others along the center line is governed by its tolerance. The distance must also be governed by a tolerance normal to the nominal center line.

The amount by which holes can be off the nominal center line must be limited. If this tolerance is $\pm S/2$ the allowable locations of centers of each hole, except the one from which the others are located, are within or on rectangles having sides T and S, as shown in Fig. 4. If the pattern for the other part is superimposed on this with the "master" hole centers and the nominal center lines coincident, the rectangles will also coincide. Then the distance between centers of corresponding holes in the two parts, in the worst combination of tolerances, called e, must equal the diagonal of the rectangles:

$$e = \sqrt{T^2 + S^2} \qquad (4)$$

Tolerance S should not be specified smaller than tolerance T. Assuming $S = k\,T$ where $k \geqq 1$, Equation (4) becomes

$$e = T\sqrt{1 + k^2} \qquad (4a)$$

Using Equation (2)

$$T = \frac{A}{\sqrt{1+k^2}} \qquad (5)$$

Equation (5) gives the tolerance on locating dimensions in terms of the allowance. This tolerance should be as large as possible, which means k as small as possible, or $k = 1$. Therefore, S is specified equal to T, and the basic formula becomes:

$$T = \frac{A}{\sqrt{2}} \qquad (5a)$$

or
$$T = 0.7A \qquad (5b)$$

The importance of specifying $S = T$ is shown by the fact that if S were 3 times T, the value of T would be less than half that given by Equation (5*b*).

In the general case the holes may be of different size, and allowances and tolerances may be different at different places in the hole pattern. If the hole in one part of Fig. 2 is a different diameter than the hole in the other parts, D_H and D'_H being the least diameters:

$$A' = D'_H - D_B \qquad (1a)$$

Then

$$A + D_B + A' = \tfrac{1}{2}D_H + e + \tfrac{1}{2}D'_H$$

or, rearranging and using Equations (1) and (1*a*),

$$e = \tfrac{1}{2}(A + A')$$

PATTERN OF TWO HOLES. Subscripts indicate serial numbers of holes.

For example,

$$\left.\begin{aligned} A_i &= D_{Hi} - D_{Bi} \\ A'_i &= D'_{Hi} - D_{Bi} \text{ where } i = 1, 2, \ldots n \end{aligned}\right\} (6)$$

Also,

$$e_i = \tfrac{1}{2}(A_i + A'_i) \text{ where } i = 1, 2, \ldots n \qquad (7)$$

The derivation of Equation (3), when changed accordingly, becomes:

$$C + \frac{T}{2} = e_1 + C - \frac{T'}{2} + e_2$$

$$e_1 + e_2 = \tfrac{1}{2}(T + T')$$

or, using Equation (7)

$$T + T' = A_1 + A'_1 + A_2 + A'_2 \qquad (8)$$

This equation gives some choice of

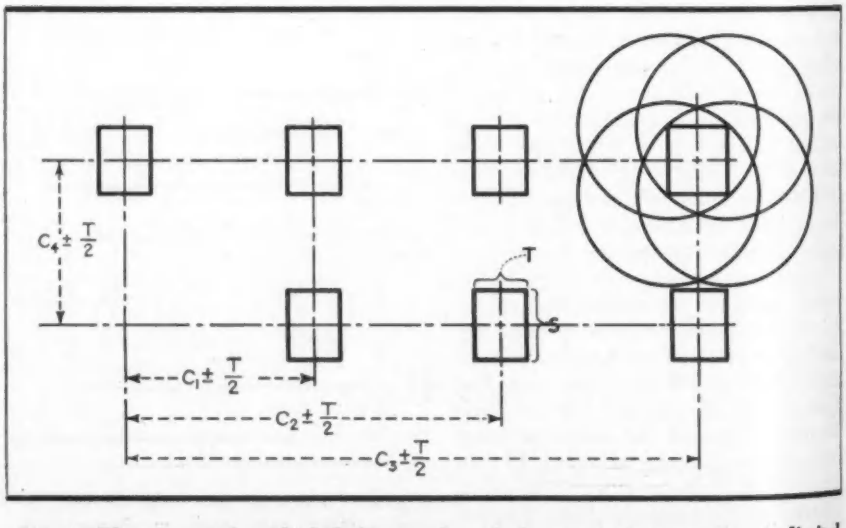


Fig. 4—The amount by which holes can be off the nominal center line is limited by a tolerance ± S/2. Rectangles inclose magnified areas in which hole centers must be located if the specified tolerances are maintained in production.

Tolerances on Locating Dimensions for Hole Centers When Interchangeability Is Required

Tolerances are based on hole diameters being the same as those of drills. The slightly greater diameter of the hole produced serves as slightly increased clearance to facilitate insertion of bolts. Blanks in drill size column indicate reamed-holes. Army-Navy Aircraft bolt numbers 3 to 16 indicate 16ths of an inch dia. hexagon head bolts; numbers 21 to 36 are clevis bolts.

Drill Size	Drill Dia.	Center Distance Tolerances, Plus or Minus: Two Holes	More than Two Holes
For AN3 bolts, diameter 0.189 in., tolerance +0.000, −0.0025			
..	0.191	±0.002	D.N.U.*
..	0.195	0.006	±0.002
9	0.196	0.007	0.002
8	0.199	0.010	0.003
7	0.201	0.012	0.004
..	0.203	0.014	0.005
6	0.204	0.015	0.005
4	0.209	0.020	0.007
2	0.221	0.030	0.010
A	0.234	.....	0.015
D	0.246	.....	0.020
E	0.250	0.06	.....
G	0.261	.....	0.025
J	0.277	0.09	0.030
For AN4 bolts, diameter 0.249 in., tolerance +0.000, −0.003			
..	0.251	±0.002	D.N.U.
..	0.255	0.006	±0.002
F	0.257	0.008	0.002
G	0.261	0.012	0.004
..	0.263	0.014	0.005
H	0.266	0.017	0.006
I	0.272	0.020	0.008
J	0.277	0.025	0.010
K	0.281	0.030	.....
M	0.295	.....	0.015
O	0.316	0.06	0.020
P	0.323	.....	0.025
R	0.339	0.09	0.030
For AN5 bolts, diameter 0.3115 in., tolerance +0.000, −0.003			
..	0.314	±0.002	D.N.U.
O	0.316	0.004	D.N.U.
..	0.318	0.006	±0.002
P	0.323	0.010	0.004
..	0.326	0.014	0.005
Q	0.332	0.020	0.007
R	0.339	0.025	0.010
S	0.348	0.030	0.012
T	0.358	.....	0.015
U	0.368	.....	0.020
V	0.377	0.06	.....
W	0.386	.....	0.025
X	0.397	0.09	0.030
For AN6 bolts, diameter, 0.374 in., tolerance +0.000, −0.003			
..	0.3765	±0.002	D.N.U.
V	0.377	0.003	D.N.U.
..	0.380	0.006	±0.002
W	0.386	0.010	0.004
..	0.388	0.014	0.005
X	0.397	0.020	0.008
Y	0.404	0.030	0.010
27/64	0.422	.....	0.015
7/16	0.438	0.06	0.020
29/64	0.453	.....	0.025
15/32	0.469	0.09	0.030
For AN7 bolts, diameter 0.4365 in., tolerance +0.000, −0.0035			
..	0.439	±0.002	D.N.U.
..	0.443	0.006	±0.002
..	0.451	0.014	0.005
29/64	0.453	0.015	0.005
15/32	0.469	0.030	0.010
31/64	0.484	.....	0.015
1/2	0.500	0.06	0.020
33/64	0.516	.....	0.025
17/32	0.531	0.09	0.030
For AN8 bolts, diameter 0.499 in., tolerance +0.000, −0.0035			
..	0.502	±0.003	D.N.U.
..	0.505	0.006	±0.002
..	0.513	0.014	0.005
33/64	0.516	0.015	0.005
17/32	0.531	0.030	0.010
35/64	0.547	.....	0.015
9/16	0.563	0.06	0.020
37/64	0.578	.....	0.025
19/32	0.594	0.09	0.030
For AN9 bolts, diameter 0.5615 in., tolerance +0.000, −0.004			
..	0.564	±0.003	D.N.U.
..	0.567	0.006	±0.002
..	0.575	0.014	0.005
37/64	0.578	0.015	0.005
19/32	0.594	0.030	0.010
39/64	0.609	.....	0.015
5/8	0.625	0.06	0.020
41/64	0.641	.....	0.025
21/32	0.656	0.09	0.030
For AN10 bolts, diameter 0.624 in., tolerance +0.000, −0.004			
..	0.627	±0.003	D.N.U.
..	0.630	0.006	±0.002
..	0.638	0.014	0.005
41/64	0.641	0.015	0.005
21/32	0.656	0.030	0.010
43/64	0.672	.....	0.015
11/16	0.688	0.06	0.020
45/64	0.703	.....	0.025
23/32	0.719	0.09	0.030
For AN12 bolts, diameter 0.749 in., tolerance +0.000, −0.0045			
..	0.752	±0.003	D.N.U.
..	0.755	0.006	±0.002
..	0.763	0.014	0.005
49/64	0.766	0.015	0.005
24/32	0.781	0.030	0.010
51/64	0.797	.....	0.015
13/16	0.813	0.06	0.020
53/64	0.828	.....	0.025
27/32	0.844	0.09	0.030
For AN14 bolts, diameter 0.874 in., tolerance +0.000, −0.005			
..	0.877	±0.003	D.N.U.
..	0.880	0.006	±0.002
..	0.888	0.014	0.005
57/64	0.891	0.015	0.005
29/32	0.906	0.030	0.010
59/64	0.922	.....	0.015
15/16	0.934	0.06	0.020
61/64	0.953	.....	0.025
31/32	0.969	0.09	0.030
For AN16 bolts, diameter 0.999 in., tolerance +0.000, −0.0055			
..	1.002	±0.003	D.N.U.
..	1.005	0.006	±0.002
..	1.013	0.014	0.005
1 1/32	1.031	0.030	0.010
1 1/16	1.063	0.06	0.020
1 3/32	1.094	0.09	0.030
For AN21 bolts, diameter 0.136 in., tolerance +0.000, −0.002			
..	0.138	±0.002	D.N.U.
28	0.1405	0.004	D.N.U.
..	0.142	0.006	±0.002
27	0.144	0.008	0.002
26	0.147	0.010	0.003
..	0.150	0.014	0.005
24	0.152	0.015	0.005
22	0.157	0.020	0.007
18	0.1695	0.030	0.010
16	0.177	.....	0.015
9	0.196	0.06	0.020
4	0.209	.....	0.025
1	0.228	0.09	0.030
For AN22 bolts, diameter 0.162 in., tolerance +0.000, −0.002			
..	0.164	±0.002	D.N.U.
..	0.168	0.006	±0.002
18	0.1695	0.007	0.002
17	0.173	0.010	0.003
..	0.176	0.014	0.005
16	0.177	0.015	0.005
14	0.182	0.020	0.007
10	0.1935	0.030	0.010
6	0.204	.....	0.015
2	0.221	0.06	0.020
A	0.234	.....	0.025
E	0.250	0.09	0.020
For AN23 bolts, diameter 0.186 in., tolerance +0.000, −0.002			
..	0.188	±0.002	D.N.U.
..	0.192	0.006	±0.002
10	0.1935	0.007	0.002
9	0.196	0.010	0.003
8	0.199	0.013	0.004
..	0.200	0.014	0.005
7	0.201	0.015	0.005
5	0.206	0.020	0.007
3	0.213	0.025	0.010
2	0.221	0.030	0.012
1	0.228	.....	0.015
D	0.246	0.06	0.020
F	0.257	.....	0.025
J	0.277	0.09	0.030

For AN24 to AN 36 clevis bolts, use tolerances for corresponding size in bolts AN4 to AN16 inclusive.

*** D.N.U. means do not use this size hole for this bolt because tolerances are not practicable. Use the next smaller.**

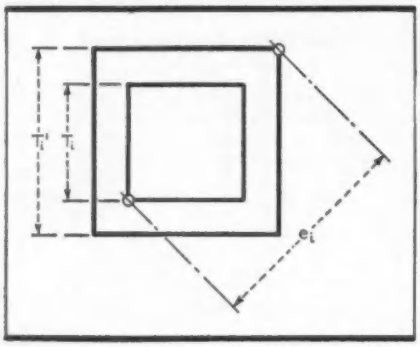


Fig. 5—When two holes, one in each plate, are of different size, tolerances may differ. Maximum mismatch, e_i, for two square areas is a simple function of the two tolerances.

tolerances between T and T' since only their sum is given, not values of each.

PATTERN OF MORE THAN TWO HOLES. If the rectangles of Fig. 4 are squares, as in Fig. 5, T_i is the tolerance for the one part and T_i' for the other. Subscripts indicate, for example, one tolerance T_2 for the C_2 dimension and a different tolerance T_3 for the C_3 dimension. It is obvious from Fig. 5 that:

$$e_i = \frac{(T_i + T_i')}{\sqrt{2}} \text{ where } i = 1, 2 \ldots n \quad (9)$$

Combining Equations (7) and (9):

$$T_i + T_i' = \frac{(A_i + A_i')}{\sqrt{2}} \quad (10)$$

where $i = 1, 2, \ldots n$

If $A_1 = A_2 = A_3 = \ldots = A_n$

and $A'_i = A_i$ for $i = 1, 2, \ldots n$

the T'_2 are all equal and Equation (10) reduces to the special Equation (5*a*).

Equation (10) is more general than Equation (5*a*) in two ways. The first is when different values are used for the two parts that are bolted together. For example, if "anchor nuts" or blind mounting nuts are attached to the back of one of the parts, the effect is to make the allowance zero.

Equation (10) then becomes

$$T + T' = \frac{A}{\sqrt{2}}$$

If T and T' are made equal, then

$$T = \frac{A}{2\sqrt{2}} \quad (5c)$$

Compare this with Equation (5*a*)

If a floating type blind mounting nut, or "basket nut" were used, the amount of float determines the effective allowance for the part to which the basket nuts are attached. If F denotes the effective allowance determined by the float, and if equal tolerances are desired on the locating dimensions for each part, then, approximately,

$$T = 0.7\frac{A + F}{2} \quad (5a)$$

In other words, instead of using the allowance on the bolt-hole fits, use the average of the allowance for the one part and the "float allowance" from the basket nuts on the other part.

The second way in which Equation (10) is more general than (5*a*) is in providing different allowances for different holes in the pattern. Fig. 6 illustrates this more clearly than Fig. 4 because the T's have subscripts and the holes are numbered according to the subscripts on the dimensions locating them. This leaves number zero for the "master" hole from which the others are located.

The master holes in each of the two parts are assumed to have their centers coincident when the parts are being bolted together. It is obvious then that the allowance for these holes need be only large enough to allow the bolt to slide in, say 0.001 or 0.002 in. The mathematical analysis confirms this. This fact is often useful in structural design because a small allowance in one hole may make the whole joint stronger in shear.

However, it may also be desirable to make a tighter fit in at least one other hole in addition to No. 0, as in Fig. 6. If the allowance is to be 0.028 in. so that the tolerance may be ±0.010 in. and if hole No. 5–3 is to have a tighter fit, say 0.006 in. allowance, this can be accomplished by locating this one hole to ±0.002 in. It is, of course, less expensive to hold one hole to ±0.002 and the rest to ±0.010 than to hold all the holes to close dimensional tolerances.

The majority of applications, however, do not need the more general formulas. Tables have been prepared for the commonest applications. They give, for various bolt sizes, the tolerances needed on locating dimensions of the holes for various hole sizes. Tables are based on the following:

1. Dimensioning as in Fig. 4.
2. Allowance same for all holes except "master" holes.
3. Allowance same in both parts for same hole.
4. Tolerance allowed for the holes to be off the line is the same as that specified for location along the line.

Where no drill size is listed, a special drill or reamer is required, probably the latter, because when the allowance, or least clearance, is that small it is usually necessary to limit the greatest tolerance also for structural reasons, and the close tolerance on the diameter is needed. In many instances of the larger tolerances in the table, the figures were rounded off by reducing 1 to 3 thousandths. For example ±0.023 appears in the table as ±0.020.

When anchor nuts are used, all tolerances in the table are divided by two, as indicated by Equation (5*c*). The drill size applies to the part without the anchor nuts.

As an example in using the table, assume an attachment consisting of four bolts through two plates. Bolt diameters are $0.249\ ^{+.000}_{-.003}$ and structural analysis requires a maximum clearance of 0.020 or a little less. The nearest listing is a reamed hole $0.263\ ^{+.001}_{-.000}$ in. in diameter, which gives a maximum clearance of 0.264 — 0.246 = 0.018 inch. Tolerance on the locating dimen-

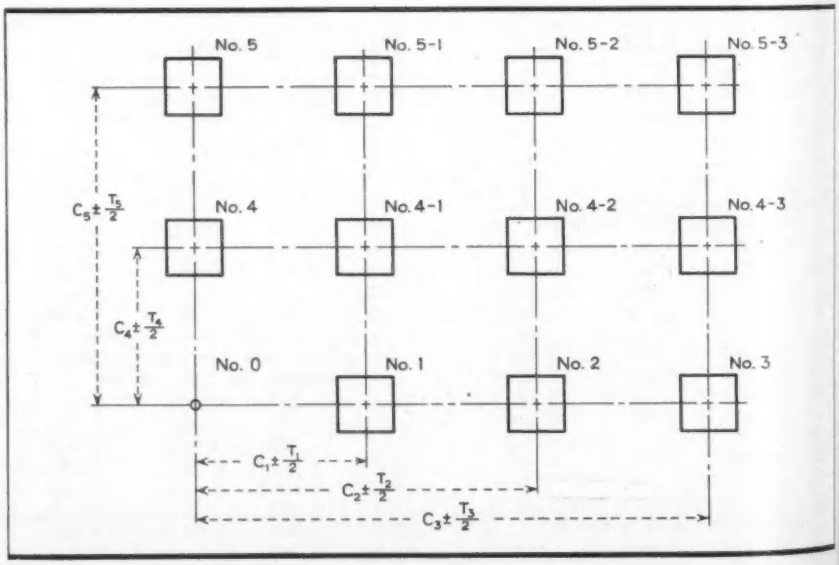


Fig. 6—System of numbering and subscripts by which to provide for different allowances for different holes in a pattern.

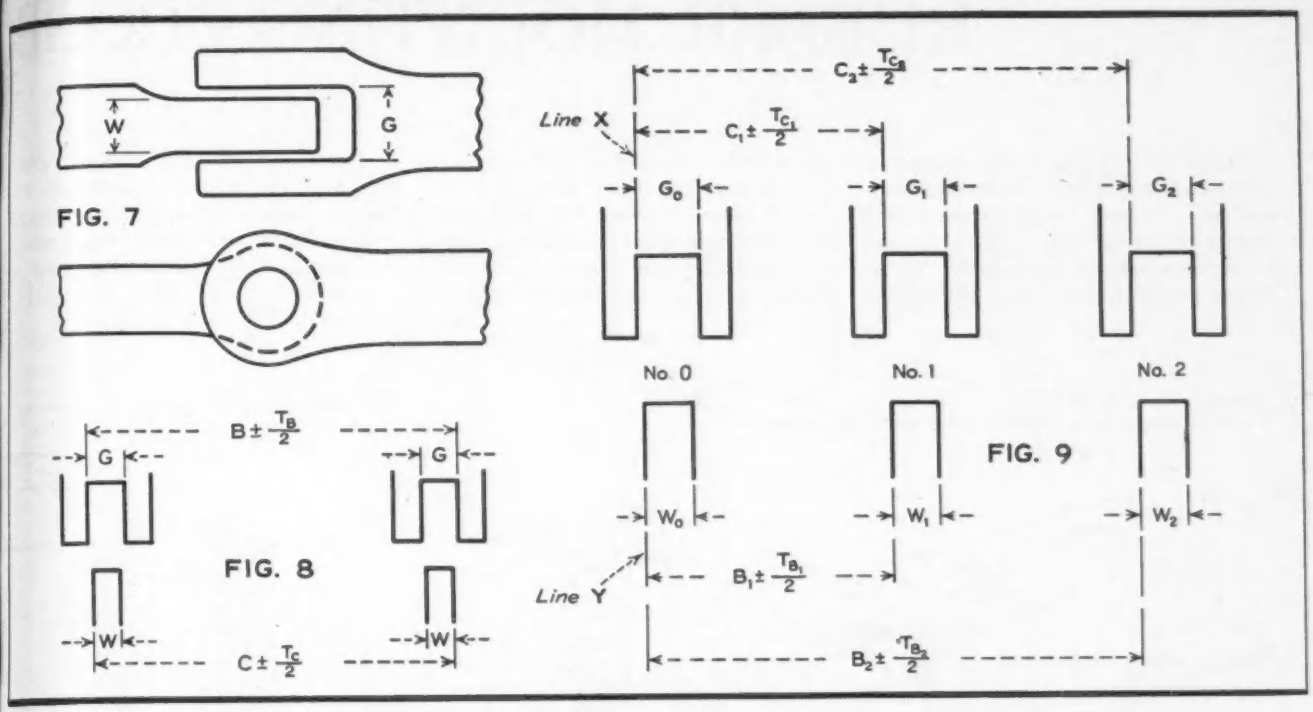


Fig. 7—Clevis-type attachment and dimensional notation. Fig. 8—Two-clevis attachment pattern and notation for derivation of formulas. Fig. 9—Three-clevis attachment pattern and notation for derivation of formulas.

sions, corresponding to 0.263 is ± 0.005.

If structural considerations were not so critical, a No. *M* drill could be used, and the tolerance on the locating dimensions could be ± 0.015 inch.

Clevis-Type Attachments

Notation in Figs. 7, 8, and 9 is:

G = least width of gap in clevis; nominal gap with positive unilateral tolerance.
W = greatest width of tongue; nominal width with negative unilateral tolerance.
P = least play between tongue and clevis.
B and C are used for locating dimensions for tongue and clevises.
T = tolerance.
Nominal subscripts are used to distinguish one clevis attachment from another.

By definition $P = G - W$ (11)

EXAMPLE 1: Two clevis attachments

In Fig. 8: $B - \frac{T_B}{2} = C + \frac{T_C}{2}$

or $B - C = \frac{T_B + T_C}{2}$ (12)

and $C - \frac{T_C}{2} - 2W = B + \frac{T_B}{2} - 2G$

or $B - C + \frac{T_B + T_C}{2} = 2(G - W)$

Using Equations (12) and (11)

$T_B + T_C = 2P$ (13)

Often $T_B = T_C = T$. Then:

$B - C = T$ (12*a*)

$T = P$ (13*a*)

In designing clevises, the least play P that is structurally allowable is determined first. Tolerances are found from Equations (13*a*) or (13). If Equation (13) is used, the sum of T_B and T_C is apportioned between the two tolerances, according to the relative difficulty of holding the two dimensions B and C. Then one of the dimensions B and C is determined and the other computed from Equation (12).

EXAMPLE 2: More than two clevises:

Attachments are numbered as in Fig. 9, the one from which the others are dimensioned being No. zero. Locating dimensions B and C carry subscripts that correspond with the number of attachments. Equation (11) becomes:

$P_i = G_i - W_i$ where $i = 0, 1, 2 \ldots n$ (11*a*)

if $n + 1$ is the number of attachments.

In order to include all combinations of tolerances, it must be assumed that the surfaces from which the other attachments are located, lines X and Y in Fig. 9, are brought together. This condition is analogous to that imposed upon the matching of hole patterns of more than two holes, where the centers of the holes from which the others are located must be coincident.

In Fig. 9:

$B_i - \frac{T_{Bi}}{2} = C_i + \frac{T_{Ci}}{2}$ where $i = 1, 2, \ldots n$

or $B_i - C_i = \frac{T_{Bi} + T_{Ci}}{2}$ (14)

Also,

$B_i + \frac{T_{Bi}}{2} + W_i = C_i - \frac{T_{Ci}}{2} + G_i$

where $i = 1, 2 \ldots n$

or $B_i - C_i + \frac{T_{Bi} + T_{Ci}}{2} = G_i - W_i$

Using quations (11*a*) and (14)

$T_{Bi} + T_{Ci} = P_i$ where $i - 1, 2, \ldots n$ (15)

If $T_{Bi} = T_{Ci} = T_i$,

$T_i = \frac{1}{2} P_i$ where $i = 1, 2 \ldots\ldots n$ (15*a*)

Since there are no locating dimensions with subscri[illegible] 0, Equation (15), which gives the tolerances in terms of least play, does not use P_0, the least play on the attachment from which the others are located. Therefore, the play on this one attachment can be very small without affecting the tolerances on the locating dimensions. This is also evident intuitively, because lines X and Y in Fig. 9 are aligned. This fact is useful structurally because one attachment can have a tight fit to take up sidewise play between two components.

Maximum Clearance

Only least clearance was considered for hole patterns and clevises as given by Equations (1) and (11) since it is least clearance that affects tolerances on locating dimensions. However, maximum clearance is usually important structurally, and has to be considered. Formulas use the following notation:

S = maximum clearance
T_G = Positive unilateral tolerance on G
T_W = negative unilateral tolerance on W
T_{DH} = positive unilateral tolerance on D_H
T_{DB} = negative unilateral tolrance on D_B

Then, for bolt holes

$S = (D_H + T_{DH}) - (D_B - T_{DB})$

for clevises

$S = (G + T_G) - (W - T_W)$

or, using Equations (1) and (11)

for bolt holes $S = A + T_{DH} + T_{DB}$

for clevises $S = P + T_G + T_W$

RUBBER MOUNTINGS COPE

Typical rubber mountings suggest a variety of applications and attaching methods for loads ranging from a few ounces to several tons, with a similar range of vibration frequencies. These designs, which include shear mountings, compression mountings, and combination mountings, were developed by United States Rubber Co. and Lord Manufacturing Co. Load limits, deflections, and vibration frequencies depend not only upon size and shape of the rubber, but upon the properties of rubber used and temperature of operation. Although rubber in compression carries greater loads per unit volume, shear mountings deflect more and are used in a wider range of forced frequencies.

Load limits, 4 to 420 lb.
Vibration frequencies, 1,000 to 3,000 cycles per min.

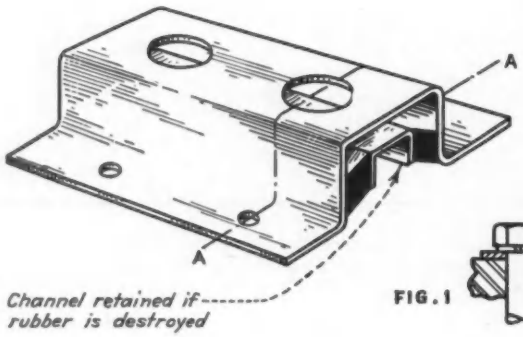
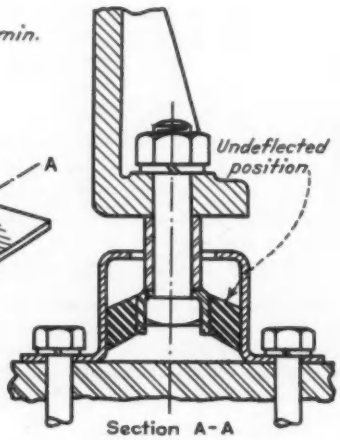
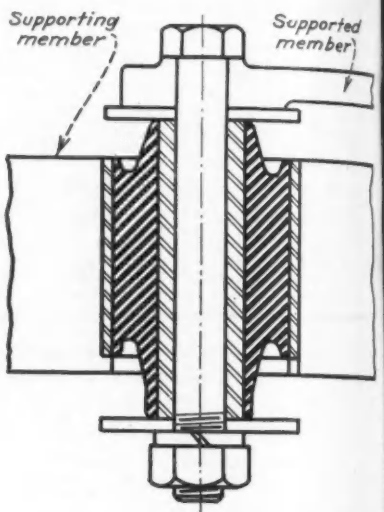


FIG. 1



Section A-A



Tube form mounting
Wide range of sizes and load capacities

Load limits, 40 to 900 lb.
Vibration frequencies 850 to 2,000 cycles per min.

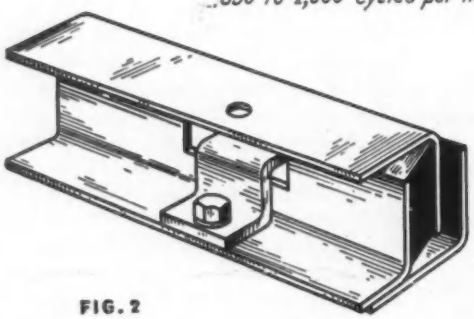
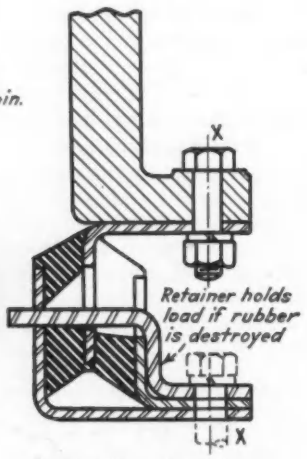


FIG. 2



Section through retainer

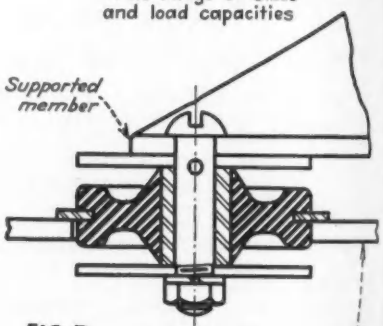


FIG. 3

Plate form mounting

Supporting member

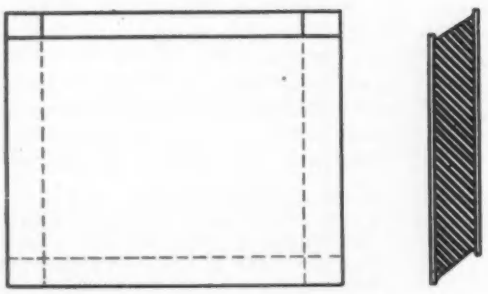


FIG. 4

Load limits 160 to 3,000 lb. in shear
Frequency, 500 to 2,000 cycles per minute
Dimensions - 13 x 9½ x 1⅞ and 9 x 17½ x 2⅛ in.

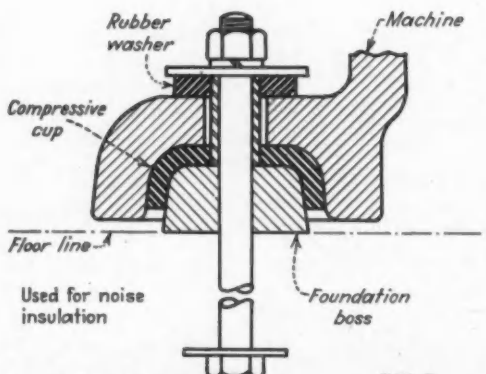


FIG. 5

Load limits 50 to 1,200 lb. vertical
15 to 400 lb. side pressure
Diameters - 2⅛ to 5⅛ in. O.D.

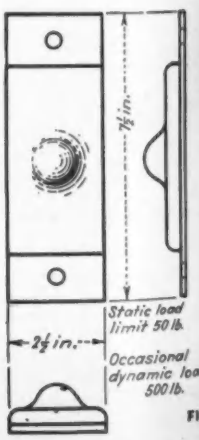


FIG. 6

WITH VIBRATION, SHOCK

Load limits,
30 to 185 lb. in shear on A axis
200 to 320 lb. in compression on B axis
40 to 70 lb. in compression on C axis
Frequency 850 to 2,500 cycles per minute

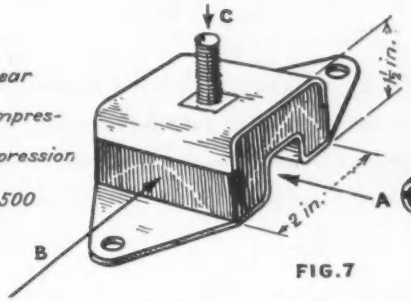
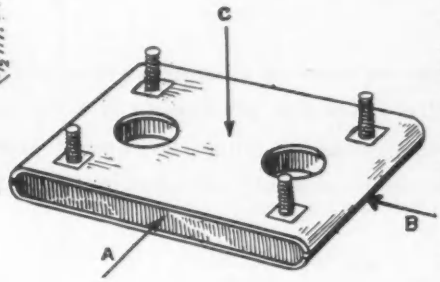


FIG. 7



Load limits,
70 to 475 lb. in shear on A axis
85 to 470 lb. in shear on B axis
Not recommended for compression unless vibrations are more than 4,500 cycles per minute
Frequency, 1,500 to 4,000 cycles per min.
Dimensions
$4 \times 4\frac{1}{2} \times \frac{3}{4}$ and $3\frac{13}{16} \times 3\frac{1}{4} \times \frac{3}{4}$ in.

Load limits 30 to 370 lb, in shear
160 to 1,050 lb, in compression
Frequency 650 to 2,500

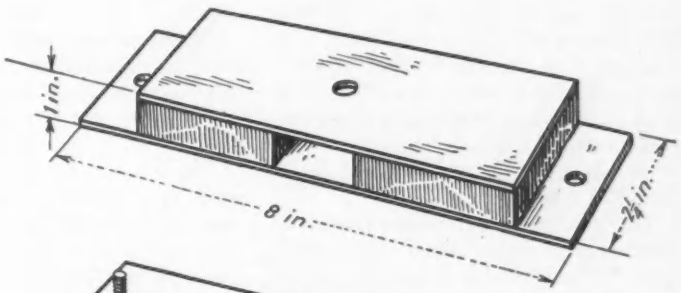
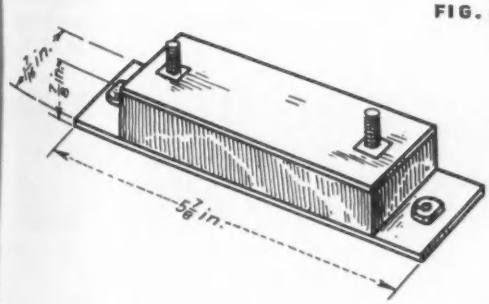
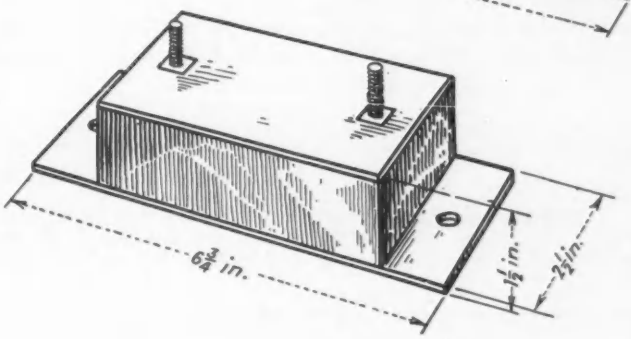


FIG. 8





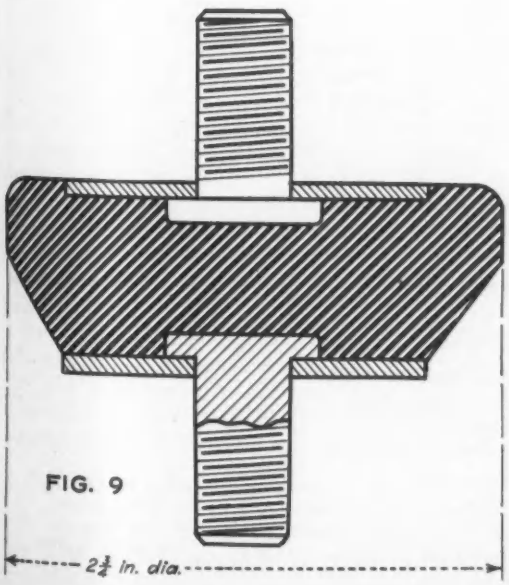


FIG. 9

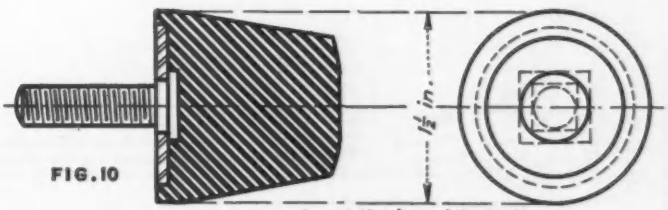


FIG. 10

Load limits, 4 to 56 lb.
Frequency, 1,100 to 3,000 cycles per min.

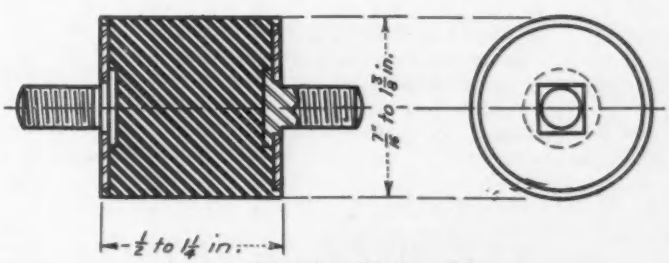


FIG. 11

Load limits, 0.5 to 33 lb. in shear
2 to 76 lb. in compression
Frequency, 700 to 3,000 cycles per min.

Graphic Cost Analyses Guide Selection of Casting Method

Effect of quantity on costs of casting various parts by different methods is shown graphically and detailed costs for producing the finished part are tabulated. By means of such analyses for various quantities of typical designs, little time is required to determine the most economical method for production of any specified quantity of a given part.

SELECTION of various methods of producing parts should not be based on snap judgment. Often it is believed that methods requiring high tooling costs are economical only when the number of pieces to be made is in the order of thousands. Cost analyses and experience often show, however, that the method involving the greatest total tooling costs is cheaper to use even when only a few hundred pieces are to be made.

Cost analyses are time-consuming and their expense adds to the cost of production. It is not unusual to find that the expense of making a detailed cost analysis is greater than the money saved in production costs as a result of the survey. Hence it is highly desirable to have cost data and graphs by means of which costs for different production methods can be compared quickly, thus enabling the engineering department to determine at little expense whether or not it is worth while to investigate further.

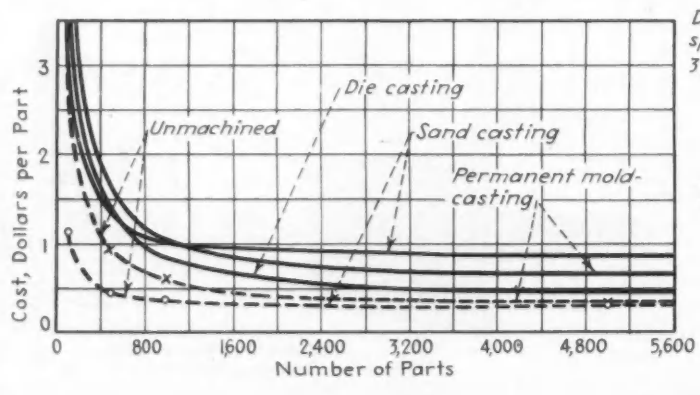
The El Segundo engineering department of Douglas Aircraft Company has compiled a wealth of cost data on various parts made by different methods. Some of that data is presented here. Although the charts and cost figures in the form as presented here have been in use for only a short time, highly gratifying results have been obtained.

A preliminary analysis of relative costs for a new item can be made by comparison with the established cost figures on a similar item. Adjustments are made by estimating. With the detailed cost figures given on the charts and tabulations, such estimates become quite accurate.

Detailed cost figures and graphs can also be used to indicate possible economies through design simplification. The cost of jigs and fixtures, machine set-up time and machining operations being

DOOR LATCH SUPPORT

Comparative costs of production by three casting methods before and after machining.



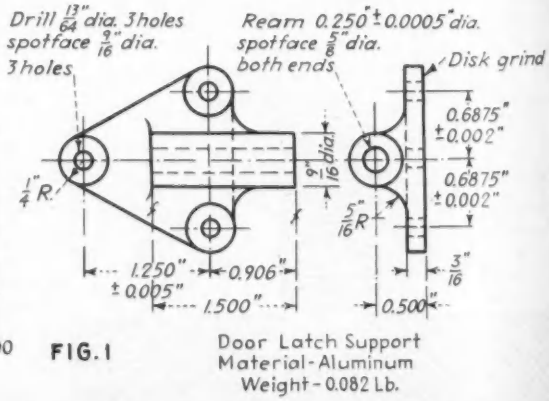


FIG. 1

Door Latch Support
Material - Aluminum
Weight - 0.082 Lb.

FABRICATION USING DIE-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

		Dollar Cost per piece
1. Fixed Tooling Cost		
One die	$475.00	
2. Material Cost		
Part purchased, No machining required		0.33
3. Set-up Cost – None		0
4. Machining Cost – None		0
5. Miscellaneous Cost		
One Shop Order Per Release, $5.00		0.050

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $513.00 or $5.13 per piece; 500 pieces $665.00 or $1.33 per piece; 1,000 pieces $858.00 or $0.86 per piece; 5,000 pieces $2,375 or $0.48 per piece.

FABRICATION USING SAND-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PIECES

		Dollars Cost per piece
1. Fixed Tooling Cost		
One pattern	$ 85.00	
One drill jig	175.00	
Total Cost	260.00	
2. Material Cost		
Cost per lb., $1.07		
Rough weight, 0.12 lb.		
Min. price per casting		0.24
3. Machine Set-up Cost		
$4.46 per 100 pieces		0.0446
4. Machining Cost per piece		
Drill, long hole	$0.0840	
Ream, long hole	0.0595	
Spotface, long hole, at 2 ends	0.0560	
Drill, 3 holes	0.0840	
Spotface, 3 holes	0.0840	
Disk Grind Base	0.0420	
Burr	0.0595	
	0.4690	
		0.47
5. Miscellaneous Cost		
1 Shop Order per release, $5.00		0.05

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $340.46 or $3.41 per piece; 500 pieces $662.30 or $1.33 per piece; 1,000 pieces $1,064.60 or $1.07 per piece; 5,000 pieces $4,283 or $0.86 per piece.

FABRICATION USING PERMANENT MOLD-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

		Dollars Cost per piece
1. Fixed Tooling Cost		
1 Permanent Mold	$350.00	
1 Drill Jig	175.00	
Total Cost	525.00	
2. Material Cost		
Price per lb., $1.00		
Rough weight per piece 0.12 lb.		
Min. price per casting		0.220
3. Machine Set-up Cost		
$2.62 per 100 pieces		0.0262
4. Machining Cost		
Drill, long hole	$0.0840	
Ream, long hole	0.0595	
Drill, 3 holes	0.0840	
Burr	0.0280	
	0.2555	
		0.26
5. Miscellaneous Cost		
1 Shop Order per release, $5.00		0.050

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $580.62 or $5.81 per piece; 500 pieces $803.10 or $1.61 per piece; 1,000 pieces $1,081.20 or $1.08 per piece; 5,000 pieces $3,306.00 or $0.67 per piece.

tabulated in detail, the designer can quickly see the possible savings if certain machining operations can be eliminated. Frequently the need for high dimensional accuracy can be avoided by simple design modifications that in no way impair the functioning of the part.

Great care should be taken before specifying a fabrication method that involves high tool charges, especially when the relative costs are not much different. Even a slight change in design, if found necessary after the dies or other tooling have been finished, might require further great tooling expense. Hence no die-casting, forging and other production methods involving large tooling costs should be specified until the design has been sufficiently stabilized.

The case studies presented here are only a few typical ones of the many that have been made by Douglas Aircraft Company at the El Segundo Engineering Department.

Machining Costs

In order to emphasize the factors that contribute to the cost of the finished part, the foundry costs of the sand-castings and permanent mold-castings are shown graphically by the dotted lines. These curves show the great effect of machining cost on the cost of the finished part. Obviously if the part could be designed to greater tolerances throughout it might be possible to eliminate both the cost of the drill jig, $175.00, the $4.46 set-up cost and the $47.00 machining cost. Under those conditions the part made as a sand-casting would, regardless of quantities, always be cheaper than the cost as a die-casting, as shown by curves in Fig. 1.

Two advantages of die-cast door latch supports other than lower costs in quantities of more than 500 pieces result from purchasing finished parts. These are first that no Douglas man-hours are required and second that machines are free for other operations.

As an example of the use of such data, terminals on large knife switches are substantially the same as the door latch support analyzed above, with these differences. The length of the lug could be cast to finished dimensions, all the holes could be finished by drilling, and the holes would be located in the casting by dimples. No drill jig would be required. Besides drilling, the only machining would be disk grinding the base.

With these design modifications the cost of the finished terminal, assuming the same material, would be based on the comparable costs of the door latch support, as follows:

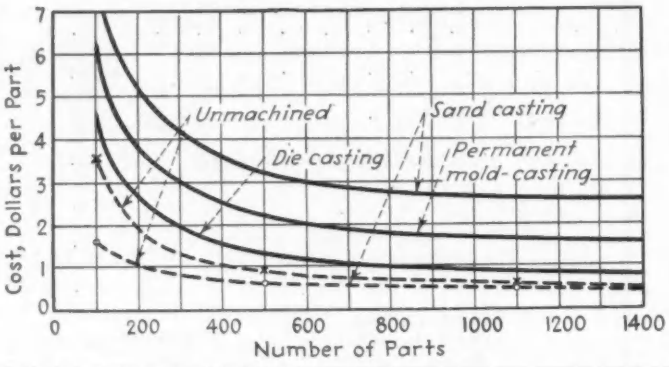
Pattern		$85.00
Cost per casting		0.24
Machining costs		
Drill long hole		$0.084
Drill 3 holes		0.084
Disk grind base		0.060
Burr		0.042
Total cost		0.270
Cost of 1,000 and 5,000 pieces in lots of 100		
Tooling cost	$85.00	$85.00
Material cost	240.00	1,200.00
Machine set-up time		
Machining cost	269.50	1,347.50
Cost of shop orders	50.00	250.00
Total cost	$644.50	$2,882.50

These figures are highly significant. One thousand finished sand-cast parts cost less than 500 die-cast parts. However, 5,000 finished sand-cast parts cost somewhat more than an equal number of die-castings. Obviously, differences in machining affect greatly the choice of fabricating method.

(continued on next page)

BEZEL FOR HYDRAULIC RESERVOIR GAGE

Comparative costs of production by three casting methods before and after machining.



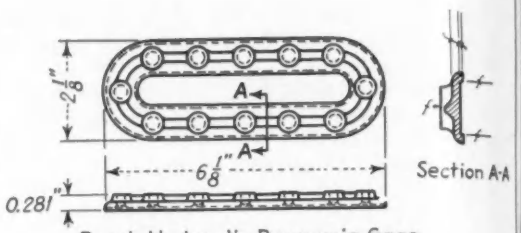


FIG. 2

FABRICATION USING PERMANENT MOLD-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
Drill jig	$227.00		
Permanent mold	325.00		
Total Cost	$552.00		
2. Material Cost			0.25
3. Machine Set-up Cost			
1 Drill set-up	$1.575		
1 Countersink set-up	1.05		
	2.625		
			0.0263
4. Machining Cost per Piece			
Drill—12 places	$0.420		
Countersink—12 places	0.336		
Burr	0.060		
	0.816		
			0.81
5. Miscellaneous Cost			
1 Shop Order per release, $5.00			0.05

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $665.63 or $6.66 per piece; 500 pieces $1,120.15 or $2.24 per piece; 1,000 pieces $1,688.30 or $1.69 per piece.

FABRICATION USING SAND-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
Pattern	$125.00		
Drill jig	227.50		
Mill fixture	262.50		
Total Cost	= $615.00		
2. Material Cost			0.30
3. Machine Set-up Cost			
4 Mill set-ups	$6.125		
1 Drill set-up	1.575		
1 Countersink set-up	1.050		
	8.75		
			0.0875
4. Machining Cost per Piece			
Drill	$0.420		
Countersink	0.336		
Face mill base	0.140		
Face mill bosses	0.140		
End mill base	0.233		
End mill top	0.234		
Burr	0.117		
	1.620		
			1.62
5. Miscellaneous Cost			
1 Shop Order per release, $5.00			0.05

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $820.75 or $8.21 per piece; 500 pieces $1,643.75 or $3.29 per piece; 1,000 pieces $2,672.50 or $2.67 per piece.

FABRICATION USING DIE-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

		Dollar Cost per piece
1. Fixed Tooling Cost		
Die Cost = $409.63		
2. Material Cost		0.45
3. Set-up Cost – None		
4. Machining Cost – None		
5. Miscellaneous Cost		
1 Shop Order Per Release = $5.00	0.05	

Total and unit costs for quantities in lots of 100 pieces are: 100 pieces $459.63 or $4.59 per piece, 500 pieces $659.63 or $1.32 per piece; 1,000 pieces $909.63 or $0.91 per piece

Die-casting is the obvious choice of method for fabrication of the bezel for a hydraulic reservoir gage since for all quantities of 100 or more it is by far the least expensive, as shown by Fig. 2. The great effect that machining operations have upon the cost of both sand-castings and permanent mold-castings is shown by the dotted curves of Fig. 2.

The same advantages of purchasing finished die-cast bezels are obtained as in using die-cast door latch supports: no Douglas man-power and machines are used. Comparing permanent mold casting with sand-casting, the former has an advantage of $63 in tooling costs alone without considering machining costs less than half as great.

BOTTOM ASSEMBLY OF HYDRAULIC RESERVOIR

Comparative costs of production by two casting methods before and after machining.

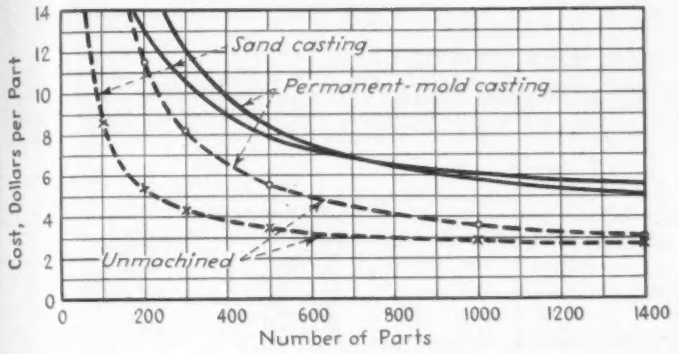
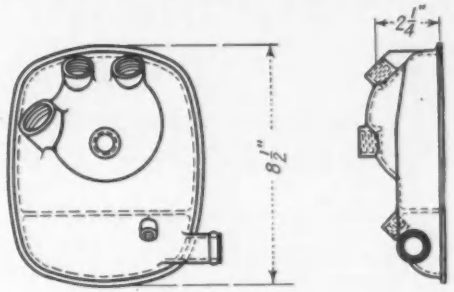


FIG. 3



Hydraulic Reservoir
Material - Aluminum

FABRICATION USING PERMANENT MOLD-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
Permanent mold		$1,975	
2 Drill jigs		700	
Total Cost		2,675	
2. Material Cost			
Rough weight	1.50 lb.		
Cost per lb.	$1.00		
Total cost per casting			1.50
3. Machine Set-up Cost			
Drill jig		$3.50	
Tapping machine		3.50	
		7.00	
Cost per piece			0.07
4. Machining Cost per piece			
Drill		0.3535	
Countersink		0.2345	
Counterbore		0.1890	
Tapper ream		0.0595	
Tapping		0.3570	
Spotface		0.2345	
		1.4280	
			1.43
5. Miscellaneous Cost			
1 Shop Order per release, $10.00			0.10

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $2,984.50 or $29.85 per piece; 500 pieces $4,222.50 or $8.45 per piece; 1,000 pieces $5,770.00 or $5.77 per piece.

FABRICATION USING SAND-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 100 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
1 pattern – metal		$ 650.00	
Form tool		25.00	
Excello fixture		350.00	
2 Drill jigs		700.00	
Total Cost		$1,725.00	
2. Material Cost			
Rough weight	1.94 lb.		
Weight cost	= $1.07 per lb.		
Total cost per casting			2.075
3. Machine Set-up Cost			
Drill jigs		$5.25	
Grinding fixture		3.50	
Tapping machine		1.75	
Excello boring mill		7.00	
		17.50	
			0.175
4. Machining Cost per piece			
Drill		$0.525	
Countersink		0.233	
Counterbore		0.189	
Taper ream		0.060	
Tapping		0.357	
Spotface		0.233	
Disk grind base		0.203	
Turn hose nipple		0.175	
		1.975	
			1.98
5. Miscellaneous Cost			
1 Shop Order Per Release			0.10

Total and unit costs for various quantities in lots of 100 pieces are: 100 pieces $2,158.00 or $21.58 per piece; 500 pieces $3,890.00 or $7.78 per piece; 1,000 pieces $6,055.00 or $6.06 per piece.

The most economical fabrication method usually becomes obvious from plotted curves and usually depends entirely upon the number of parts to be made. However, the most economical method is not always the one properly chosen. The cost analysis for the hydraulic reservoir shown in Fig. 3 illustrates this fact.

Despite the fact that this hydraulic reservoir costs at least $2 more as a permanent mold-casting than as a sand-casting in quantities of 250, the former process would probably be the logical choice for production for this quantity because of the other factors involved.

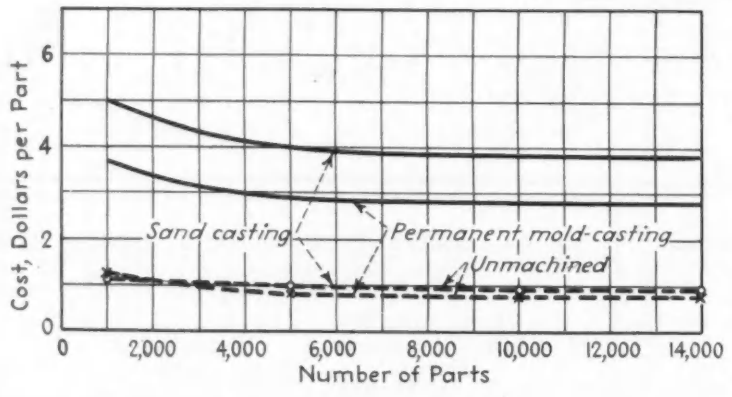
In this instance the weight saved by use of permanent mold-casting, evaluated at $10 per lb. because it is an aircraft part, amounts to an equivalent money saving of $2.80 a part, which would

more than compensate for the higher actual money cost of the permanent mold-casting. The relative lack of porosity of the permanent mold-casting also makes its use desirable in order to insure a leak-proof casting. A further consideration by Douglas engineers is a saving in Douglas man-hours.

The difference in costs between sand- and permanent mold-castings is greater if production tooling and machining costs are not taken into account, as indicated in Fig. 3 by the dotted curves. The high permanent mold cost is caused by the necessity for having six core pulls in five different directions and a loose piece in the mold to form the undercut under the hose nipple.

WING-FLAP GEAR BOX HOUSING

Comparative costs of production by two casting methods before and after machining.



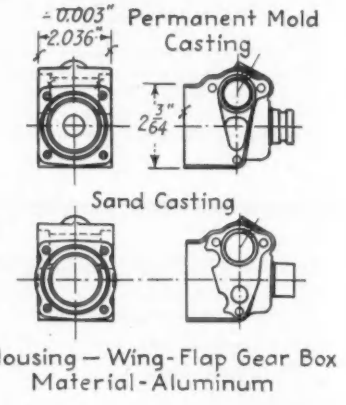


FIG. 4

Housing — Wing-Flap Gear Box
Material - Aluminum

FABRICATION USING PERMANENT MOLD-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 1,000 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
1 Drill jig	$350.00		
1 Lathe fixture	175.00		
Mold cost	= 475.00		
Total Tooling Cost	= $1,000.00		
2. Material Cost			
0.578 lb. at $1.10 per lb. =			0.635
3. Machine Set-up Cost			0.0123
4. Machining Cost per Piece			
Bore		$0.863	
Drill, Ream, Countersink, Tap		1.050	
Burr		0.117	
		2.030	2.03
5. Miscellaneous Cost			
1 Shop Order per release, $10.00			0.01

Total and unit costs for various quantities in lots of 1,000 pieces are: 1,000 pieces $3,687.25 or $3.69 per piece; 5,000 pieces $14,436.25 or $2.89 per piece; 10,000 pieces $27,872.50 or $2.79 per piece; 50,000 pieces $135,362.50 or $2.71 per piece.

FABRICATION USING SAND-CASTING

COSTS BASED ON PRODUCTION IN LOTS OR RELEASES OF 1,000 PARTS

			Dollars Cost per piece
1. Fixed Tooling Cost			
1 Drill jig	$472.50		
1 Lathe fixture	175.00		
1 Milling fixture	350.00		
1 Pattern	245.00		
Total Tooling Cost	$1,242.50		
2. Material Cost			
0.743 lb. at $1.10 per lb. =			0.817
3. Machine Set-up Cost			0.0175
4. Machining Cost per Piece			
Bore		0.864	
Turn		0.116	
Drill, Ream, Countersink, Tap		1.341	
Mill		0.350	
Burr		0.233	
		2.904	2.90
5. Miscellaneous Cost			
1 Shop Order per Release, $10.00			0.01

Total and unit costs for various quantities in lots of 1,000 pieces are: 1,000 pieces $4,991.00 or $4.99 per piece; 5,000 pieces $19,985.00 or $4.00 per piece; 10,000 pieces $38,733.50 or $3.87 per piece; 50,000 pieces $188,697.50 or $3.77 per piece.

Permanent mold-casting is the obvious choice in Fig. 4 of the fabrication of the housing for the wing-flap gear box on account of savings in time, money, and weight. A much larger quantity is required than of the other parts described, which accounts for the fact that the curves are nearly parallel through the whole range. Costs, as cast, for this half-pound housing are less than 15 cents a part, in favor of permanent mold-casting more than 3,000 pieces. Machining operations add roughly $2 to the cost of the permanent mold-casting and $3 to that of the sand-cast housing in the larger quantities.

Other production methods will be compared in the August issue.

Lightweight Plywood Tubing Is Strong Structural Material

Fabricating methods which play an important part in the sturdiness of tubing are explianed. Several war products are described to illustrate what can be done with plywood tubing.

LAMINATED wood veneers bonded together on a mandrel to form tubing result in a structural material that is exceptionally strong for its weight. War uses for plywood tubing show that it can be satisfactorily used for structures, containers, pipe lines and some fabricated products where light weight and relatively high strength are important.

Strength characteristics of plywood tubing are determined mostly by the method of laminating the veneers and glue, rather than the kinds of wood and glue used. There are several alternatives in the wrapping method and choice of the one to be used is best left to the judgment of the manufacturer because not only the method of wrapping but the specie of veneer affect the characteristics of the tubing.

Chemical properties depend on the characteristics of the glue formulation. Synthetic resin glues have helped tremendously in the development of tubing with outstanding chemical resistance through impregnation. Both inner and outer surfaces of tubing can be coated with glue.

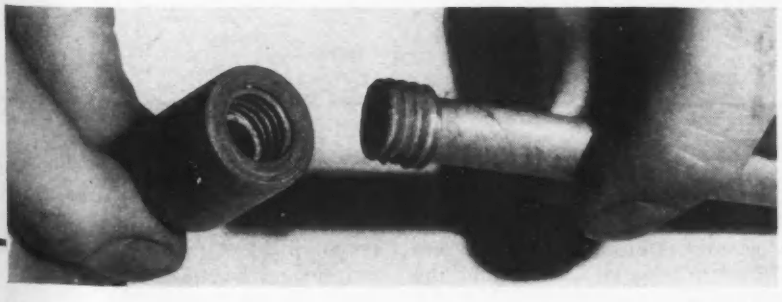
Urea-formaldehyde glue is the usual bonding agent for Plytube, but phenolic resins also can be used. These glues impart resistance to moisture, flame, corrosive action, natural decay of the wood and heat. The tubing is dimensionally stable under temperatures as high as the resin will stand. This ranges from approximately 230 to 450 deg. F. for phenolic and to about 300 deg. F. for ureas.

In making Plytube the Plymold Corporation builds up veneers by automatic wrapping in such a manner that stress or strain in any direction acts on the total columnar grain fibers of the veneer layers. Angle of wrap with respect to the long axis of the tube to a large extent determines the strength characteristics. With the inside layer spiral wound, for instance, compressive strength is emphasized. Wrapping combinations emphasize different strength characteristics by varying the angles at which the layers are wound. There are combinations developed especially for radial crushing, torsion, columnar compression, flexural and tensile strengths. All of these can be combined so that the fiber stress in all directions approximates that of the maximum columnar strength of the wood.

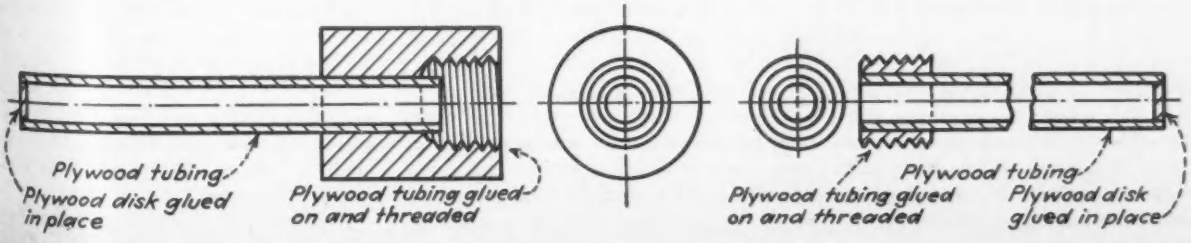
Plytube is outstanding in that its specific gravity is about 0.85. Impregnation of the wood before wrapping is undesirable because of the increased weight that would result. However, ends of tubes can be impregnated to facilitate threading. Because thin veneers are used on the surface there is ordinarily little or no crumbling when the tube is threaded.

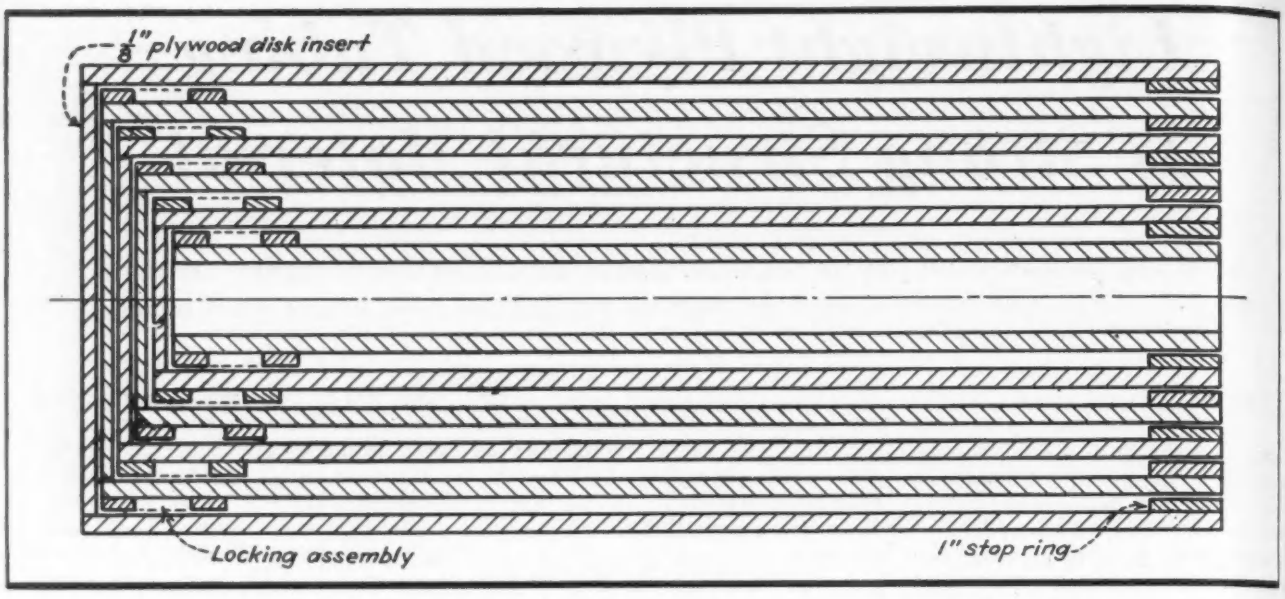
Electrical characteristics depend more on the wood than the resin because the resin content of the tube is only about 15 percent and the dielectric constant of

Tripod made of plywood tubing weighs 2.5 lb. and replaces a 10-lb. brass unit. Legs telescope and have friction lock to hold them at desired height. A 25-ft. telescoping aerial mast has the same weight and also uses friction locking.

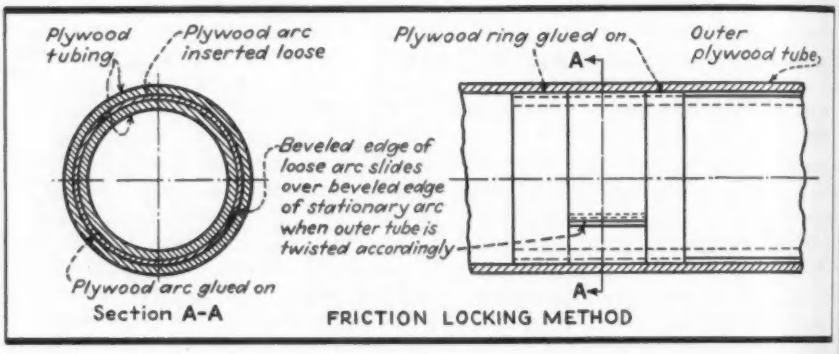


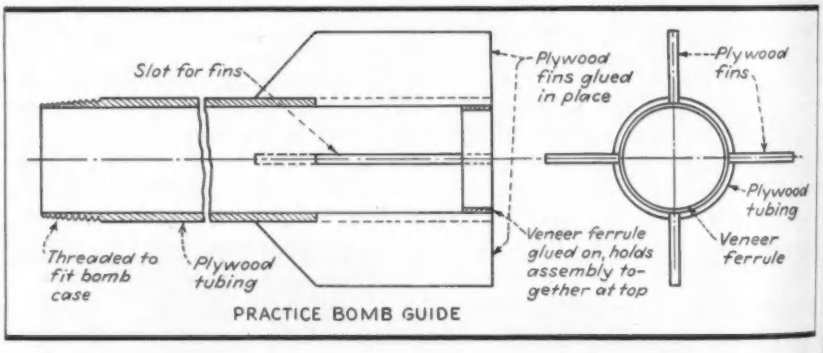
DELAY DETONATOR CASE


Plywood tubing
Plywood disk glued in place
Plywood tubing glued on and threaded
Plywood tubing glued on and threaded
Plywood tubing
Plywood disk glued in place



Short mast for use in life rafts collapses to 14 in. and makes use of a friction lock to hold it at desired height. Tube sizes range from ½ in. I. D. for the smallest to 1¾ in. I. D. for the largest.


Plywood tubing
Plywood arc inserted loose
Plywood ring glued on
A
Outer plywood tube
Beveled edge of loose arc slides over beveled edge of stationary arc when outer tube is twisted accordingly
Plywood arc glued on
Section A-A
A
FRICTION LOCKING METHOD



the resin is approximately the same as that of the wood. Conductivity is, therefore, low. Another unusual feature is that metal foil can be used as the inside or outside layer or in place of one of the intermediate veneer layers.

For decorative purposes, the outer veneer can be made of special woods and grains. In this way, decorative effects comparable to those on veneered furniture and veneer can be obtained.

Plytube is now available with inside diameters ¼ to 36 in., wall thicknesses of 0.030 to 1.000 in. and in any reasonable length. Tolerances on inside diameter of relatively small tubing are held to minus 0 and plus 0.010 in. and tolerances on wall thickness are plus or minus 0.005 in. These limits on size are for standard tubing. Smaller diameter tubing can be made if necessary and larger tubing awaits a demand for it.

In addition to round tubing, square and oval sections are possible. Tapered and conical sections have also been made, one such project calling for a taper ranging from 8 to 24 in. over 3 ft. of length.

Specification data for plywood tubing cover a wide range as test results depend on the types of veneer, glue, diameter, wall thickness and construction used. The values in the accompanying table are intended to serve merely as a guide.

Uses of plywood tubing vary. Structural columns and antenna towers naturally take advantage of lightness and strength. Electrical conduit is made with a very thin wall. Pump lines and reservoirs are constructed from tubing developed especially to withstand radial pressures. Land mines, radar equipment and compass tripods are made from the tubing because of the non-metallic properties of the material. Plywood tubing with flexural strength emphasized has been used in place of bamboo and aluminum poles.

Fabrication of Plytube offers several alternatives. Tubes can be nested and glued to make a composite structure. Short lengths of tubing built up by disks of plywood or ordinary wood and surrounded by another piece of tubing make up a cellular construction. Ends of tubes can be plugged with disks and dowels. Synthetic resin or non-waterproof glues are often used to hold pieces together, but threading is used on many assemblies that have to be taken apart. Metal fittings also can be used. Motor shafts, incidentally, have the veneers wrapped so that the twist is against the grain to resist torque.

Threading is done with ordinary pipe dies and cutting can be done with regular wood-cutting machinery. Straight threaded joints are made by the combination of threads and flanges or by male and female unions. Flanged joints are made by gluing flat plywood or plain wood to the tube.

Products of plywood tubing have been limited so far to war uses. These

include sectional life raft oars, aircraft wing spars, demolition floats, parachute flare casings, carrying cases, ski pole shafts, ski pole snow rings, dowel replacements, aircraft antennas, aircraft fuel tank inner supports, cargo plane fire extinguisher supports, telescoping tent poles, ship railings, camouflage netting poles, percussion caps, toboggan shafts, grease reservoirs, air-borne litter poles, airplane seat supports, bazooka tubes and seaplane fending-off poles.

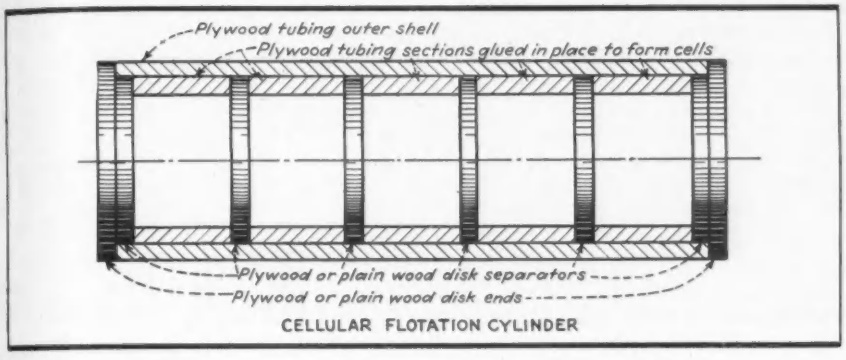
Masts and tripods lend themselves readily to the advantages of Plytube. They can be made in sections of any desired length and can be carried easily because they are light. Collapsible masts 5½, 25, 50 and 90 ft. long when extended are in use by the armed forces. Maximum head loads of 8,000 lb. are mounted on the 50- and 90-ft. masts. Development of a 150-ft. mast that will take an 8,000-lb. load is now under way.

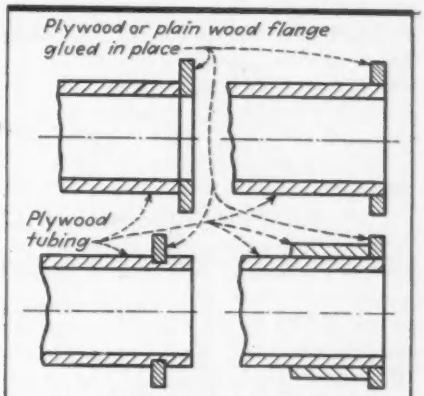
To hold the tubes of the shorter masts and the tripods in place the friction locking assembly illustrated was developed. The longer masts are assembled by having a Plytube stop ring mounted a short distance from the end of each tube. The end of the next smaller size tube then rests on this ring.

The shorter masts and tripod are nested by unlocking and pushing the sections together. The longer masts are nested by disassembling, reversing the tubes and inserting one inside the other.

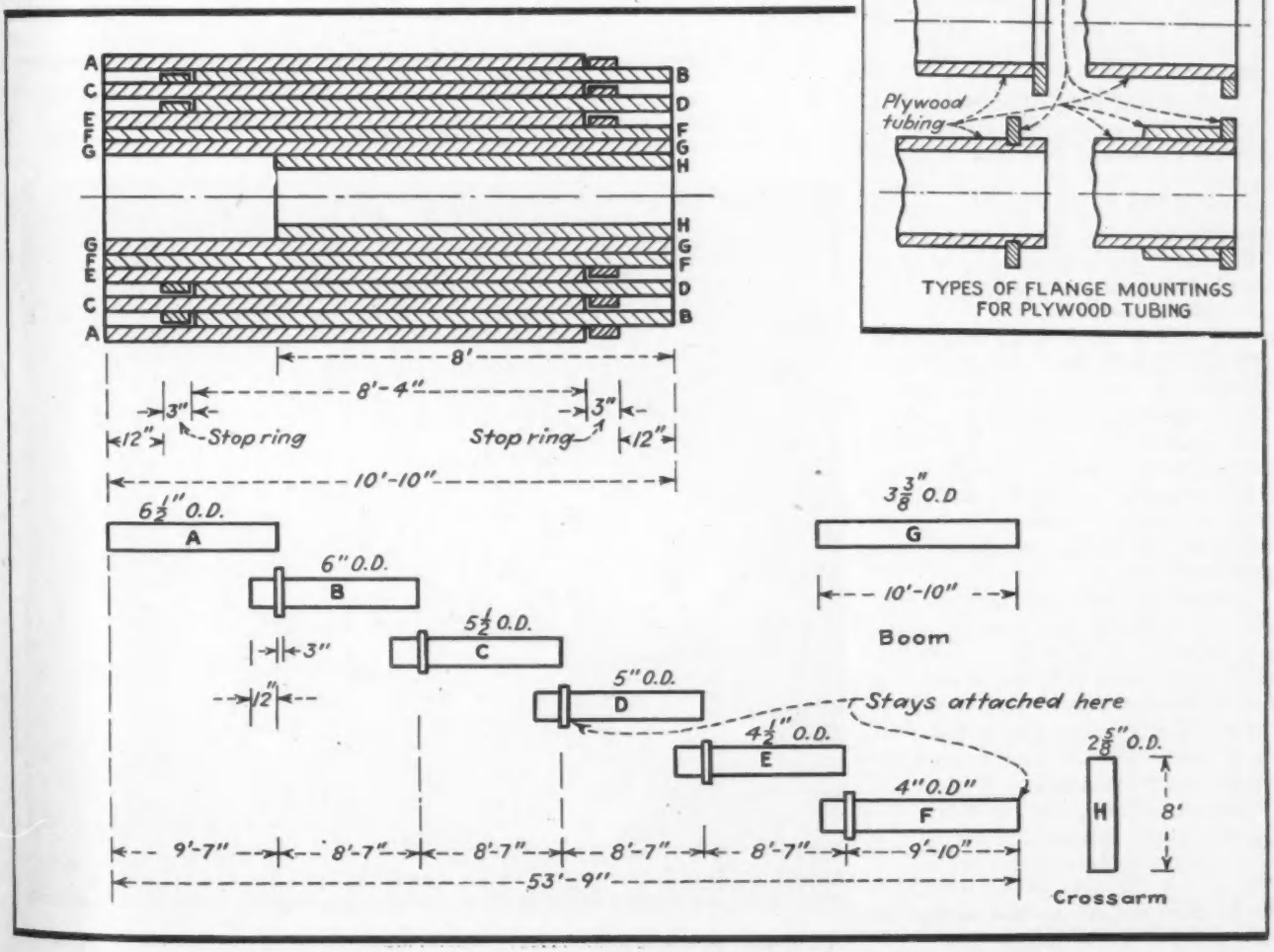
AVERAGE STRENGTH CHARACTERISTICS OF PLYTUBE, LB. PER SQ. IN.

Ultimate Compressive Strength.	11,500
Ultimate Tensile Strength.....	11,000
Compressive Proportionality Limit	6,000
Tensional Proportionality Limit	6,000
Deflection, Young's modulus...	1,500,000
Axial Expansion, Young's modulus	1,700,000
Axial Compression, Young's modulus	1,400,000
Buckling Failure, Young's modulus, Euler's formula.....	11,000


Plywood tubing outer shell
Plywood tubing sections glued in place to form cells
Plywood or plain wood disk separators
Plywood or plain wood disk ends
CELLULAR FLOTATION CYLINDER



TYPES OF FLANGE MOUNTINGS FOR PLYWOOD TUBING



Collapsible 50 ft. antenna mast of the Signal Corps can be assembled and erected by one man in ½ hr. Mast is assembled on the ground and all fittings and guys attached. Then it is pulled up by means of the boom and a block-and-tackle. Weight of the mast proper is 35 lb., but fittings and guy wires weigh 49 lb. All sections, including the boom and crossarm, nest into a compact unit.

INDUSTRY AND SOCIETIES

NEWS • READER LETTERS • MEETINGS

Airplane Models Tested at 700 M.P.H. In New Boeing Wind Tunnel

THE FASTEST among the large wind tunnels in the country was dedicated recently by Boeing Aircraft Company in honor of Edmund T. Allen, famed Boeing test pilot and scientist who lost his life last year in an airplane test accident. The new wind tunnel is a self-contained research unit, including model design, construction facilities, computing laboratories, and testing quarters. It has many innovations not found in any other single tunnel and is "streamlined" for efficiency of operation.

Model planes with wingspreads up to 11 ft., or full-scale airplane sections of the same maximum size, can be tested in the new tunnel. Wind speeds ranging up to 700 m.p.h. can be produced through the 12-ft. wide test sections. All controls are centralized in a panel board before the test section where observations are made.

The flow of air is created by a propeller-like fan 24 ft. in diameter. This fan consists of 16 laminated spruce blades mounted on the end of a 37-ft. solid steel drive-shaft 16 in. in diameter. The synchronous electric motor has a rating of 18,000 hp. and maintains a constant speed of 514 r.p.m. Speed of the fan is regulated by a magnetic coupling.

An intricate system of balances, capable of measuring with great accuracy lifts from 0.1 to 8,000 lb., records lift, drag, yaw, pitch, roll and side force. An ingenious automatic printer will, at the touch of a button, record and print on a tape all the forces acting on the model at any instant.

The tunnel bore is a complete, continuous-return structure which follows an approximately rectangular course 450 ft. long. It varies in size from 8x12 ft. in the "throat" or test section to 27½x27½ ft. at the largest part.

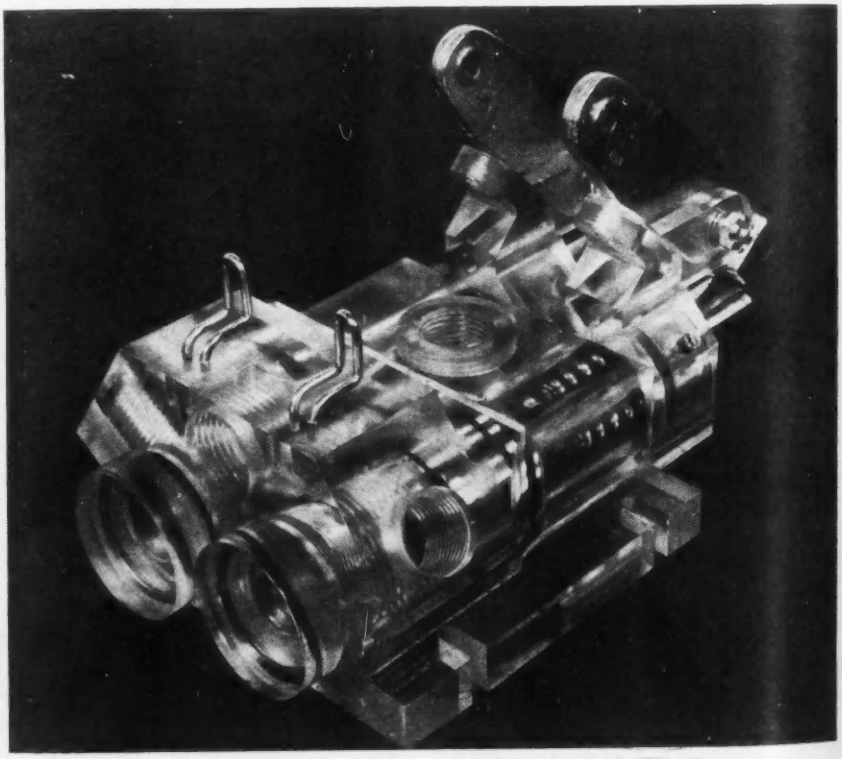
When operated at high speed the air completes the 450-ft. circuit in less than 2 sec. Inasmuch as 11 percent of the air in the wind tunnel is replaced with fresh air at every "round trip" for cooling purposes, under high-speed operation the air in the tunnel is replaced three times per minute. A pagoda-like structure, known as an interchanger tower, is located on top of the tunnel building and is used for intake and exhaust of this air.

An aerodynamics laboratory adjoins the wind tunnel, with a reception lobby and model-building shops on the first floor and engineering offices and tunnel operations room on the second. An overhead monorail system transports models from the construction shops through the operations room and into the test section. The tunnel is soundproofed throughout.

Duct Material Saves Ten Percent in Weight

MORE THAN 10 percent in weight is saved in an air duct made of fabric, asbestos and a coating of synthetic resin.

Clear Plastic Parts Used for Education



Aircraft parts made of clear Plexiglas acrylic plastic are used for visual education of workers at the Ford Willow Run bomber plant. Test models of new designs are also made of the material. Part functions and engineering fundamentals involved are easily explained in this manner. Shown here is a model of the brake valve, used on the Consolidated B-24 bomber.

Helicopter Flown Indoors at Demonstration

What is believed to be the first indoor flight in this country of a helicopter took place recently in the 65th Regiment Armory in Buffalo, N. Y., before officers, members and cadets of the Civil Air Patrol. The demonstration was arranged to show the stability and precision control of the craft and its ability to take off, maneuver and land in restricted areas. The craft used for the demonstration was built under the supervision of Arthur Young, who heads up the helicopter program for the Bell Aircraft Corporation. In this design the main rotor is rendered independent of the mast by a stabilizing device which tends to keep the rotor on a horizontal plane. Also, the rotor does not tip.

Developed by United States Rubber Company, this pliable fabric can be twisted and turned or completely deformed and returned to its original shape. Called Multiplex, it is fire resistant and can be used for either hot- or cold-air systems.

Simplicity of construction and ease of application and installation allow this material to be formed into radii and intricate angles and shapes for fittings. Patterns, and bending and fitting of joints and elbows, are unnecessary. Parts are assembled and cemented with a special synthetic-rubber compound.

The new duct is used on aircraft to heat carburetors, as cooling duct, header ducts, generator blast tubes, fire-wall seals, for cooling or heating cabins of combat and cargo ships, windshield defroster extensible hose and as machine-gun shell ejector chutes. A special system has been devised to keep bombsights at constant temperature.

Discussions and Comments from Readers

SHRINK-FIT CALCULATOR SHOWS DISCREPANCIES IN PUBLISHED DATA

To the Editor:

The Reference Book Sheet, "Shrink Fit Calculator, on page 71 in your January, 1944, issue, makes a brave showing with its needlessly elaborate diagram.

The expansion in length is assumed as proportional to the temperature rise or range, that is, $L_t = L_i\ (1 + \alpha t)$, in which L_i is the initial length, L_t the expanded length due to the rise in temperature t and α the thermal coefficient of expansion for the particular metal; or the increase Δ is $\alpha\ L_i t$—simplicity itself. The latter makes a radial chart in two quadrants; or if preferred, a nomogram in four parallel scales with one ungraduated turning scale.

However, over large temperature ranges as are used for shrinkage, for refined calculation the variation is not quite linear, as assumed, but follows the parabolic equation $L_t = L_i\ (1\ t + \alpha t + \beta t^2)$ as given in the "Handbook of Chemistry and Physics." In this for iron, $\alpha = 0.1145 \times 10^{-4}$ and $\beta = 0.071 \times 10^{-7}$, t being deg. C.

In the simplified form as in the second paragraph, α is not constant but ranges from 9.07 to 15×10^{-6} for iron, according to the point of temperature considered.

Moreover, the term "nominal diameter" really means the diameter at the initial or at some given temperature.

Furthermore, the article states the same coefficient applies to either diameters or to lengths, which we think is contrary to fact. The cubical coefficient is roughly three times the linear one.

Thus, for a cylinder of initial volume V_i

$$V_i = \frac{\pi}{4} D_i^2 L_i \qquad (1)$$

Whence $$D_i^2 = \frac{4 V_i}{\pi L_i} \qquad (2)$$

Now $$V_t = V_i (1 + 3 \alpha t) \qquad (3)$$

and $$L_t = L_i (1 + \alpha t) \qquad (4)$$

Substituting in (2)

$$D_t = \sqrt{\frac{4}{\pi} \times \frac{V_i (1 + 3 \alpha t)}{L_i (1 + \alpha t)}} \qquad (5)$$

This is not the same as $D_t = D_i\ (1 + \alpha t)$, as assumed. In the article

the assumed shrinkage is doubled, perhaps to compensate for some of the discrepancies mentioned.

—CARL P. NACHOD

To the Editor:

The discrepancy which Mr. Nachod mentions was the result of using handbook data too freely. The coefficients of expansion which I used to plot the chart were taken from the handbook, *Machinery*, 11th edition, page 1580. *Machinery's* data made no reference as to what range of temperatures the coefficients of expansion applied.

The very fact that this point has been brought out as a result of this article should create interest among the publishers of standard handbooks and technical works to simplify and correctly inform the industrial public as to the validity and range to which the published data applies.

There is, in my opinion, a definite need for simplification and standardization of the data on coefficients of expansion as applied to industrial shrink- and expansion-fit problems. As to the remark "needlessly elaborate diagram," perhaps Mr. Nachod has a simpler approach to this problem. If so, I am sure that we shall all welcome his constructive approach to the solution of the condition which I mentioned in the preceding paragraph.

In the second paragraph of his letter, Mr. Nachod's analysis of the method by which the chart was constructed is correct.

His third paragraph mentions the parabolic equation,

$$l_t = l_o (1 + \alpha t + \beta t^2)$$

where l_o = length at 0 deg. C.
l_t = length at temperature t.

This formula is found on page 1341 of *Handbook of Chemistry and Physics*, 23rd edition. The coefficients α and β for various metals cover only a relatively small temperature range. If Mr. Nachod wishes to go a little further on this problem he might note the following equation for aluminum between 0 deg. C and −200 deg. C. This equation introduces a third term. It was taken from *National Metals Handbook*, 1933, page 1031.

$$l_t = l_o [1 + (22.56t + 0.01675 t^2 - 0.00003667 t^3) 10^{-6}]$$

To continue with the discussion, where Mr. Nachod states that in the simplified form of the expansion formula that α is not constant he is quite right as I have already acknowledged, but the error appear larger than it really is because of the range chosen for the coefficients. In the case of iron it is as follows:

Temperature range	Coefficient
−190 to 17 deg. C.	9.07×10^{-6}
300 to 400 deg. C.	15.00×10^{-6}

Actually, the range of the chart is from −115 to 265 deg. F., or −81.7 to 129.4 deg. C. Reference to Fig. 1 will show an average coefficient for iron of approximately 10.1×10^{-6}. In terms of Fahrenheit this would be 5.61×10^{-6}. Using separate coefficients for the heating and the cooling ranges we have an error in the use of the chart of 6.6 percent and 10.3 percent, respectively. This is more than I would like to have, but it is not unreasonable for shop calculations, where the chart was to be used as a quick and ready check for shrink-fit problems.

I have not had sufficient time to prepare an elaborate and meticulous report nor have I been able to find complete data on the various metals covered by the chart. I have roughly analyzed the data I found on copper and will discuss it in reference to altering the chart.

For copper we have practically a straight line increase in the coefficient from 0 deg. C. and upward. Below 0 deg. C. the coefficient drops off gradually at first and then quite rapidly after the temperature gets below −100 deg. C. Reference to Fig. 2 will show that between the range of −81.7 and 129.4 deg. C., which is the range of the shrinkage calculator, I have arbitrarily selected 21.1 deg. C. (70 deg. F.) as the average starting point or room temperature. The average coefficients then become 15.8×10^{-6} for the cooling range and 16.9×10^{-6} for the heating range. By converting these coefficients to apply to Fahrenheit temperatures and plotting on the shrink-fit calculator, I would have instead of a single line three lines which would form a band. The outside lines giving the coefficients for the heating and cooling ranges and the center line would give the coefficient for work straddling these two

Electronic Heating Sets Twist in Rayon Cord

By placing packages of rayon tire cord in a high-frequency electrical field the twist of the cord is uniformly set in a few minutes by Industrial Rayon Corporation. Control of moisture content of the cord is facilitated by wrapping cones in moisture-proof paper before processing them. The cones shown here weigh 18 lb. each and are used in the weftless method of tire construction. High-frequency heating units for this installation and those at other Industrial Rayon plants were supplied by Girdler Corporation's Thermex Division (PRODUCT ENGINEERING, June, 1943, page 40). Each unit can handle several thousand pounds of packaged tire cord in a 24-hr. period. The process, patent applications on which have been assigned to Industrial Rayon, is also being used by B. F. Goodrich Company under license to produce rayon tire cord, with high-frequency heating units supplied by Radio Corporation of America.

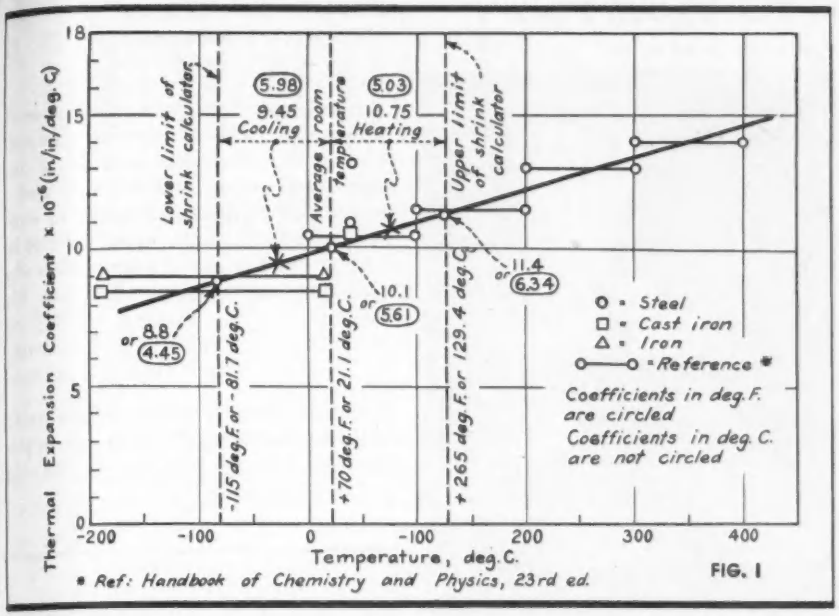


FIG. 1

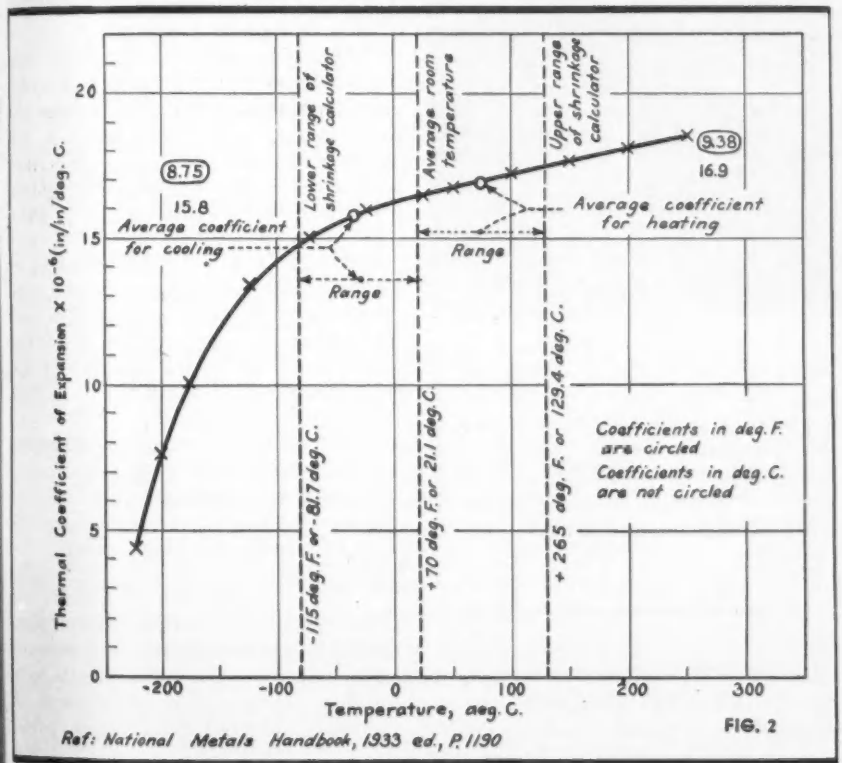

Thermal Coefficient of Expansion X 10⁻⁶ (in/in/deg. C.
Temperature, deg. C.
Lower range of shrinkage calculator
Average room temperature
Upper range of shrinkage calculator
8.75
15.8
Average coefficient for cooling
Range
9.38
16.9
Average coefficient for heating
Range
-115 deg. F. or -81.7 deg. C.
+70 deg. F. or 21.1 deg. C.
+265 deg. F. or 129.4 deg. C.
Coefficients in deg. F. are circled
Coefficients in deg. C. are not circled
Ref: National Metals Handbook, 1933 ed., P. 1190

FIG. 2

ranges. The error which would then be incurred would be slight.

Mr. Nachod stated in his letter that the same coefficient does not apply to both diameters or lengths. If the mathematical proof is completed the finished equation would appear as shown in the calculations for deviation for D_t from the volumetric expansion relationship. Using a problem and substituting the values into both forms of the equations, the final result was a difference of only a few ten-thousandths of an inch. For shrink fits in the shop we usually can not work to such limits.

"Doubling the assumed shrinkage to compensate for some of the discrepancies" is only common shop practice.

If you can recommend any authoritative source of data which will give me complete information on the expansion characteristics of the various metals, I should be glad to try to work out a satisfactory revision to the published chart. —EDWARD L. ROGERS

Derivation for D_t From the Volumetric Expansion Relationship

$$V_t - V_i = 3\,\alpha\,V_i\,t \quad (1)$$

$$V_i = \frac{\pi D_i^2 L_i}{4} \text{ and } V_t = \frac{\pi D_t L_t}{4} \quad (2)$$

Substituting (2) in (1)

$$\frac{\pi D_t^2 L_t}{4} - \frac{\pi D_i^2 L_i}{4} = \frac{3\,\alpha\,\pi D_i^2 L_i t}{4} \quad (3)$$

$$D_t^2 L_i - D_i^2 L_i = 3\,\alpha\,D_i^2 L_i t \quad (4)$$

$$D_t^2 L_t = D_i^2 L_i (1 + 3\,\alpha\,t) \quad (5)$$

$$D_t^2 = \frac{D_i^2 L_i (1 + 3\,\alpha\,t)}{L_t} \quad (6)$$

But

$$L_t = L_i (1 + \alpha\,t) \quad (7)$$

Substituting (7) in (6)

$$D_t^2 = \frac{D_i^2 L_i (1 + 3\,\alpha\,t)}{L_i (1 + \alpha\,t)} \quad (8)$$

$$D_t = \sqrt{\frac{D_i^2 (1 + 3\,\alpha\,t)}{(1 + \alpha\,t)}} \quad (9)$$

$$D_t = D_i \sqrt{\frac{1 + 3\,\alpha\,t}{1 + \alpha\,t}} \quad (10)$$

Compare (10) with this form

$$D_t = D_i (1 \pm \tfrac{n}{d}\,\alpha\,t)$$

Examples

A Given—10-in. dia. bronze bar.
Rise in temperature, 32 deg. F. to 212 deg. F.
Average coefficient (32 deg. F. to 212 deg. F.), 10×10^{-6} (Marks, p. 298).

Using Equation (10)

$$D_t = D_i \sqrt{\frac{1 + 3\,\alpha\,t}{1 + \alpha\,t}}$$

$$= 10 \sqrt{\frac{1 + (3 \times 10 \times 10^{-6} \times 180)}{1 + (10 \times 10^{-6} \times 180)}}$$

$$= 10 \sqrt{\frac{1.0054}{1.0018}}$$

$$= 10 \sqrt{1.00359}$$

$$= 10.0179 \text{ in.}$$

Using Equation (1A)

$$D_t = D_i (1 + \alpha\,t)$$

$$= 10 [1 + (10 \times 10^{-6} \times 180)]$$

$$= 10.0180 \text{ in.}$$

B Given—5-in. dia. bushing
Rise in temperature, 65 deg. F. to 265 deg. F.
Average coefficient (65 deg. F. to 265 deg. F.), 9.38×10^{-6}

Using Equation (10)

$$D_t = 5 \sqrt{\frac{1 + (3 \times 9.38 \times 10^{-6} \times 200)}{1 + 9.38 \times 10^{-6} \times 200)}}$$

$$= 5.0094 \text{ in.}$$

Using Equation (1A)

$$D_t = 5 (1 + (9.38 \times 10^{-6} \times 200)$$

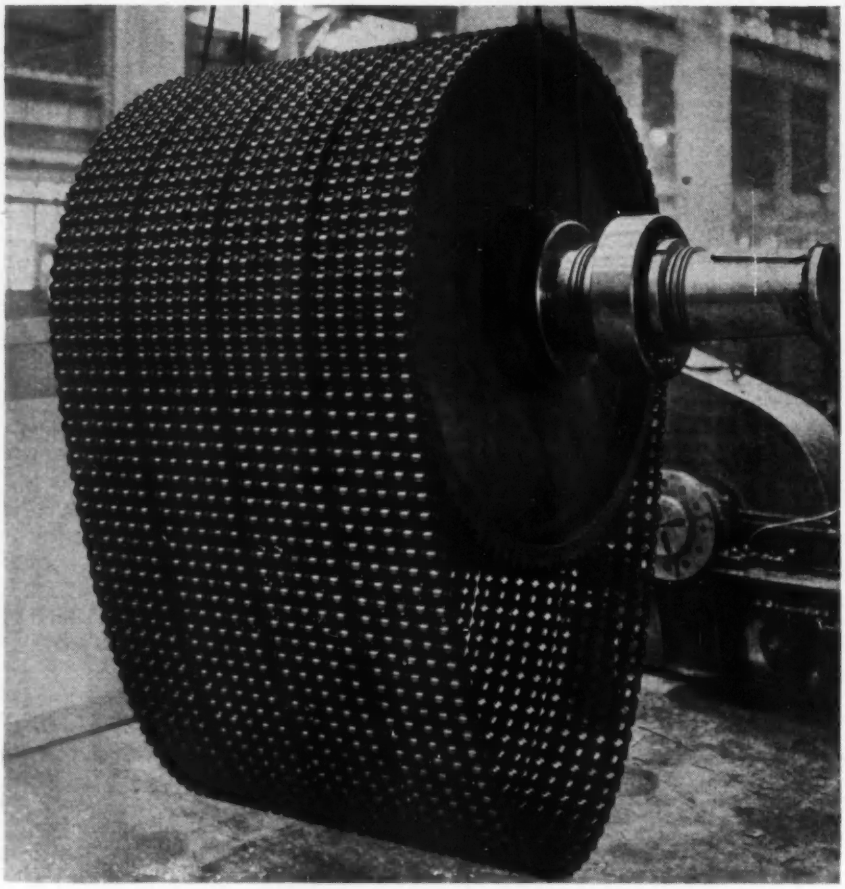
$$= (5 \times 1.001876) = 5.0095 \text{ in.}$$

SLIDE THE TIMBER AROUND THE CORNER

To the Editor:

Here is a problem the solution of which lends itself readily to the calculus and the use of logs. Two hallways, one 3 ft. wide and the other 4 ft. wide, meet at a right angle. A 12x12-in. timber that is too heavy to lift or roll is to be skidded around the corner by cutting it off square to the maximum length possible to slide around the corner. What is this maximum length? —L. W. BEAVEN

Marine Propeller Drive Has 10,000 Parts



Complete interchangeability of chains, elimination of "take-up" and perfect matching are obtained by precise specifications and new manufacturing methods developed by Morse Chain Company for marine propeller drives like this one, which contains 10,000 parts. Used in the Navy's 110-ft. harbor tugs this drive, largest application of its kind, couples two direct-reversing diesel engines to a single propeller with 3-to-1 reduction. Advantages include operating efficiency, economy and flexibility of engine-room layout.

ADDED DATA INCREASE VALUE OF SPRING CHART

To the Editor:

Mr. Carl Thumim's spring chart in your November, 1942, number, facing page 662, filled a long-felt want as it enables the designer to take into account the Wahl factor and the different stresses applying to different wire sizes, all without extra calculation. However, there is an error under $\frac{5}{32}$ O.D. opposite 0.0258 *d*. The deflection should be 0.0185 instead of 0.0815.

An addition or a box, like that shown below, directly on the chart, would prevent neglecting to apply the fatigue factor corrections. This tabulation is a condensation of Mr. Thumim's Table II with an explanation.

—HAROLD L. BLOOD, *New Design Heald Machine Company*

TENSION SPRINGS

Working Range From	0 to Full Load	½ to Full Load	¾ to Full Load
Frequent Cycles	0.40	0.45	0.60
Moderate Service	0.45	0.50	0.65
Nearly Static	....	....	0.70

COMPRESSION SPRINGS

Working Range From	0 to Full Load	½ to Full Load	¾ to Full Load
Frequent Cycles	0.50	0.60	0.80
Moderate Service	0.60	0.70	0.85
Nearly Static	....	....	0.90

DETAILS OF MACHINE SOUGHT BY READER

To the Editor:

I am in quest for teaching purposes of information concerning the Sanford-Mallory flax-breaking machine, a machine used—or at least was once used—in the flax-producing industry to separate the fiber from the wood. I should be very grateful to any of your readers who may have information concerning this machine.

—WILLIAM H. RASCHE, *Professor of Mechanism, Virginia Polytechnic Institute.*

EDITOR'S NOTE—PRODUCT ENGINEERING will be glad to publish answers to this problem from readers who may be able to help Professor Rasche.

Do You Know That—

POLYSTYRENE MOLDING POWDERS have been reduced 3 cents in price to 27 cents per lb., made possible by the huge expansion of production facilities for making monomeric styrene, important ingredient of Buna S synthetic rubber. (23)

WHEN A SECTION OF WOOD about ¼ to ½ in. thick is moistened on one end-grain face, the piece will begin to curl away from the moist side because of the force produced by the swelling of the wood fibers on the wet side. (24)

CRACKS IN CASINGS made of non-magnetic materials, such as aluminum, copper and their alloys, are found by dipping the castings in a special preparation and inspecting them under ultra-violet light. (25)

STICKERS mounted on translucent tape, so they can be run through an Addressograph machine and printed with code numbers, are used for identification of individual aircraft replacement parts. (26)

METAL COATINGS are applied to steatite insulator surfaces by fixing a layer of silver to the surface at high temperature and electroplating copper on top of the silver. (27)

BATTLESHIPS of the Iowa class which have a displacement of 45,000 tons require more than 1,500 tons of copper and copper alloys. (28)

IF HALF THE LENGTH of a wood strip 12 in. long is treated with a good water-repellent solution to resist the penetration of water that portion of the wood will remain straight, while the untreated half will curl. (29)

FREE ENTERPRISE

The Obligation of Management and Labor to Cooperate...in War...in Peace

The Invasion is on! We have unleashed our full might for military victory. We have confidence that our great strength will bring success. We are strong because we have achieved unity in mobilization and in combat.

Though victory appears assured, we cannot rest until we have done everything in our power to speed the day when death and destruction are halted.

The home front is an important factor in this time element, for the fighting power of our Armed Forces depends upon their weapons. Napoleon's army fought "on its stomach"—man against man. Eisenhower's men fight on their tonnage—tanks, artillery, machine guns, heavy bombers.

As never before in the long succession of wars, the legends of heroic deeds on the battlefronts in this world conflict will be paralleled in history by the great accomplishments on the production fronts. Along with these heroic achievements of our Armed Forces, the world will long remember the record of our production accomplishments which have made us the stongest military power in the world, as well as the arsenal of democracy.

As the conflict reaches its climax, as battles grow fiercer and more destructive, our responsibility becomes greater and more critical. We must coordinate our productive efforts with the same ingenuity and the same precision with which our Armed Forces have coordinated theirs. We dare not waste the productivity of a single man or machine in these critical days.

As our landing craft are discharging our fighting men on the beaches of Europe and the Pacific, they must not want for equipment. No interference with war production for any reason can be justified. There must be no picket lines in America!

The landing of American troops in France virtually has stopped all strikes in the United States. This is important and encouraging news because the prelude to Invasion, unfortunately, has been an epidemic of strikes. Time lost through strikes, during the first four months of 1944, was double that lost during the same period last year. April saw more strikes than any other month since Pearl Harbor, and in May the record again was broken. Here is what happened within two weeks in May:

> Nine thousand men in six Chrysler plants in Detroit were out when a jurisdictional dispute in a "soda pop" war between the American Federation of Labor teamsters and the Congress of Industrial Organizations fired their discontent.
>
> A three-day sit-down strike occurred among 950 employees in the B. H. Aircraft plant over the refusal of the company to discharge a superintendent unsatisfactory to the union.
>
> Thirteen hundred men in the Chevrolet transmission and axle plant at Saginaw struck over a no-smoking rule and a change in shift-starting time.
>
> Two thousand employees at the Browne and Sharpe Manufacturing Company walked out when a woman was hired to fill a job long held by a man.
>
> Production of penicillin, blood plasma, and other medical supplies was halted at two Detroit plants of the Parke Davis Company as 1900 employees struck for a ten-cent raise.
>
> Over 25,000 lumber workers in the Pacific Northwest struck because the War Labor Board denied their demand for a wage increase.

At the end of the third week of May, 70,000 workers in 26 plants in Detroit were idle because of strikes.

Strikes in Detroit alone reduced production as much as a moderately successful German air raid would have done. Far more important than their effect on output is the effect of strikes upon national unity and morale. To our home front and to our Armed Forces, strikes belie our pledge to back the attack with all the power at our command. Hence, strikes limit our all-out war effort.

Prompt and decisive action is needed to keep America free from strikes for the remainder of the war. Stoppages of work on the production lines cannot be condoned while lives are being lost in fighting the enemy.

Most union leaders realize this need and are preparing to impose discipline upon their members who violate the no-strike pledge. The Warehouse Division of the International Longshoremen's and Warehousemen's Union (C.I.O.) recently declared: "Strikes in this time of war are treason against the nation and betrayal of the interests of labor." A message sent by William Green to all heads of American Federation of Labor unions stated:

> "D-day is here. From now on until Hitler is finally crushed, every worker enrolled in the army of production must consider himself a part of the invasion forces of the United States and conduct himself accordingly. I call on you in the name of the American boys who are risking their lives under enemy fire to maintain uninterrupted production under any and all circumstances. Until victory is won every worker must give the same all-out service that our Armed Forces are giving on the field of battle."

Strongest of all was the appeal of R. J. Thomas, president of the United Automobile Workers, to members of his union:

> "Our union cannot survive if the nation and our soldiers believe that we are obstructing the war effort . . . there can be no such thing as legitimate picket lines . . . I appeal to our membership. If you value your union, if you want to live and serve after the war, we must restrain ourselves and our hot-headed brothers today. If we do not, there will be no union after the war."

Union officers are entitled to vigorous support from management and government in their efforts to prevent strikes. Behind many a strike is an accumulation of unsettled grievances. Managements are overworked, and many union shop stewards are new and inexperienced and do not always do their part in turning down cases which lack merit. Both of these conditions make it easy for large backlogs of unsettled grievances to pile up. A special drive to clean up unsettled cases and to prevent new accumulations of them is one way by which managements and local union officials can help shorten the war.

The government too has a contribution to make to the prevention of strikes—both through the prompt disposal of disputes and through firm action against the leaders of strikes. The National War Labor Board and the Regional Boards are disposing of over five thousand cases a month and have made an excellent record in reducing their backlogs. Nevertheless, the boards still have many old cases; and about one out of four strikes has been an effort to get action from one of the labor boards. The boards are entitled to cooperation from employers and unions in keeping down their docket. In instance after instance, cases are dumped in the lap of the board before the union and employer have made a real effort to get a meeting of minds and to work out settlements.

In the present emergency, strikes are an expression of the lack of adequate understanding and team work between unions and management. Any future great upsurge in industrial strife likewise will be due to misunderstanding. After this war this country must not go through another "1919" when the time lost from strikes reached an all-time high. With 13 million workers, or almost half of the non-salaried employees of the country, in trade unions, the power and prestige of unions is greater than ever. ***The long-run prosperity of the country requires that business and labor learn how to cooperate in supporting the policies which produce the largest possible profits and the largest possible payrolls.***

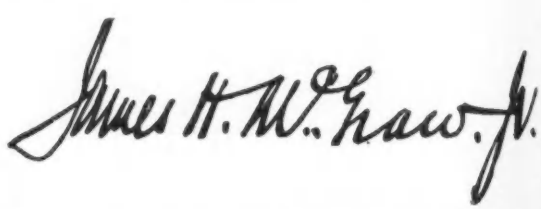
Although business is primarily interested in the largest possible profits, and labor is primarily interested in the largest possible payrolls, both objectives call for the same basic conditions. Payrolls depend upon the prospects for profits. If bad relations between business and labor or unwise public policies cause employers to take a pessimistic view of the outlook for profits, both employment and payrolls will be depressed.

Individual unions and individual employers always will have differences over wages and hours and the status of labor in particular plants or in particular occupations. Some disputes on such issues are inevitable, but resort to arbitration and calm intelligence can help greatly in avoiding strikes in the long run. Cooperation between labor and management is an economic necessity. In our kind of economy, payrolls and profits ***both depend upon the willingness and the ability of business and labor to work together in creating the conditions under which enterprise flourishes.***

The foundation for intelligent and effective cooperation must be accomplished by skillful and imaginative managers in plants throughout the country who are willing to help unions with their problems, and who are able to interest union leaders and their members in the problems of business. Union members and their leaders are keenly interested as a rule in the efforts of management to win new markets. They know that jobs depend upon the success of managements in improving the product, adding new items to the line, and, less frequently, cutting costs and prices. Employees like to be kept informed about what management is doing, what problems it is meeting, and what success it is having. Most of all, they like to have an opportunity to contribute their ideas and suggestions.

The recent epidemic of strikes should not blind us to the fact that even today there are more plants where managements and unions are on good terms than ever before in the country's history. Consider, on the one hand, the extensive and constantly growing efforts of unions to train and develop shop stewards and, on the other hand, the efforts of employers to teach foremen how to carry out the new responsibilities imposed upon them by union agreements. Unions and managements together are learning how to operate together such technical devices as time study and job evaluation. Managements which, several years ago, opposed the provision of umpires to interpret union agreements and to settle deadlocked cases today are taking the lead in suggesting such arrangements.

The war is reaching a crisis, and all groups in the country must be aware as never before of their common interests. This presents an opportunity which should be seized to lay the permanent foundations for more effective team work in American industry. Let history record that the days when Europe was being liberated also were the days when unions and employers were making unprecedented progress in preparing American industry for the return of the service men by developing policies of cooperation between business and unions. Such cooperation will help achieve a peace worthy of our efforts and our sacrifices.



President, McGraw-Hill Publishing Company, Inc.

New Books

The Fundamentals Of Electrochemistry And Electrodeposition

SAMUEL GLASSTONE. 90 *pages*, 5⅛x8 *in., red leather covers. Published by American Electroplaters' Society*, 545 *Fifth Ave., New York, N. Y. Price* $2.

Intended primarily as a simple introduction to the fundamentals of electrochemistry for the benefit of practical electroplaters, this book can be useful also to engineers whose work could be made more effective by an understanding of such principles. The book is divided into 20 chapters of four or five pages each. The subject matter of each chapter logically follows that of the preceding chapter, starting with the formation of simple ions and ending with the causes and prevention of corrosion. This final chapter, on corrosion, contains some interesting notes on the subject.

The Engineer's Manual of English

W. O. SYPHERD, ALVIN M. FOUNTAIN and SHARON BROWN. *Revised Edition.* 503 *pages*, 5x7½ *in., black clothboard* covers. *Published by Scott, Foresman & Co.*, 623 *Wabash Ave., Chicago, Ill. Price* $2.50.

One of the most notable improvements in this edition is the integration of the specimens directly with the discussions of the respective kinds of technical writing. Many of the specimens, too, have been chosen from recent publications. Also, the sections on paragraphs, sentences and words have been made into a separate chapter, giving them the emphasis they deserve. The chapter on writing for technical journals has been considerably improved. Two appendices, one on military communications and the other on public speaking, have been added.

Graphical Solutions

CHARLES O. MACKEY, 152 *pages*, 5¼ x 8½ *in., blue clothboard covers. Published by John Wiley & Sons*, 440 *Fourth Ave., New York* 16, *N. Y. Price* $2.50.

Second edition of this text, this volume expands the subject of alignment charts, explaining the use of determinants for charts with curved scales, and includes a new chapter on the fitting of empirical equations to periodic curves. The book is about equally divided among the three subjects: Sliding scales and network charts, alignment charts, and empirical equations and curves. Graphical Solutions was written to be used in presenting a college course. To a large extent it depends for its effectiveness upon classroom explanation and practice, and upon an instructor's knowledge of actual procedure in applying the methods described. The book presents practical mathematical tools for engineers, but with a minimum of instruction in their use.

Technical Data On Plastic Materials

Published by Plastic Materials Manufacturers' Association. 631 *Tower Bldg., 14th and K Sts., Washington* 5, *D. C.* 141 *pages*, 8½ x 11 *in., maroon cardboard covers. Price* $1.50.

Nature, particular merits and utility of various plastic materials are indicated and property values commonly to be expected in available forms and modifications of the various basic compositions are presented in tabular form. Eighteen types of plastics are covered, each in a separate section with a brief explanatory foreword. The three-page introduction is an excellent general discussion on material selection and parts fabrication.

The Chemistry Of Synthetic Substances

EMIL DREHER. 103 *pages*, 5⅜ x 8¼ *in., tan clothboard covers. Published by Philosophical Library, Inc.*, 15 *E. 40th St., New York, N. Y. Price* $3.

Although this book deals with the chemistry of macromolecular organic compounds it can be useful to engineers who wish to become acquainted with the chemical processes by which modern synthetic substances are produced. It is a collection of previously published essays that have been translated from German and is, apparently, notable for its completeness.

Blueprint Reading

FRED NICHOLSON AND FRED JONES. 141 *pages*, 7x9¾ *in., blue clothboard covers. Published by D. Van Nostrand Co.*, 250 *Fourth Ave., New York, N. Y. Price* $2.

Many years of experience in teaching blueprint reading led to the development of the method used in this book wherein 50 lessons and blueprints are supplemented by illustrations of tools and shop procedures with instruction relative to their use. The authors hope that by following this method students will not only learn how to read blueprints but also will be able to interpret the language of industry and will learn what must be done in the shop to accomplish the purpose of the designer.

600 New Things Coming After the War

Published by Business Bourse, 80 *W. 40th St., New York, N. Y.* 44 *pages*, 8½x11 *in., loose-leaf ring binder. Price* $7.50.

Of the new materials, methods and processes developed during the war period, 600 that will be available after the war have been chosen by the Business Bourse research staff and compiled in synopsis form. Developments in numerous fields have been brought together in this collection. Many of them are already generally known, others will be new to most people. The important thing is that these 600 developments have been listed in a single source. No indication is given of the organizations responsible for these developments.

Bulletins

Bearing Manual No. 44

Published by Bearing Manual, Inc., 9 *S. Clinton St., Chicago* 6, *Ill.* 88 *pages*, 8⅜ x 11 *in., orange cardboard covers. Price* $2.50.

Ball and roller bearings are listed according to type and series. A separate chart for each series lists the catalog numbers and dimensions for each manufacturer of that series so that interchangeability is apparent at a glance. No reference is made to load capacities and other technical data.

Recommended Standards For Single-Stage and Multi-Stage Mechanical Drive Steam Turbine

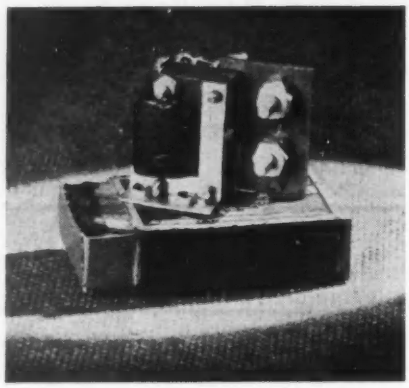
Published by National Electrical Manufacturers Association, 155 *E. 44th St., New York* 17, *N. Y.* 12 *pages*, 8 x 10½ *in., brown cardboard covers. Price* 50 *cents.*

First NEMA publication pertaining to single- and multi-stage mechanical drive steam turbines, this one provides standards for nomenclature, basis of ratings, pressure and temperature, load capacities, overspeed protective devices, relief valves for discharge connections, speed governed by external devices and standard features and accessories.

NEW MATERIALS AND PARTS

Autotransformers

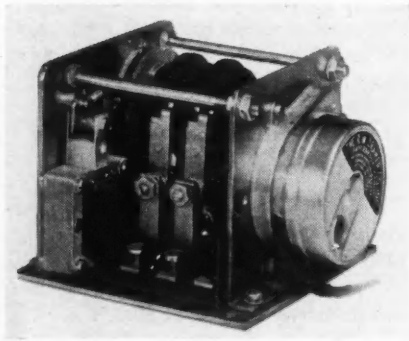
Furnishing of proper a.c. voltage for operation of 400-cycle aircraft instruments, such as the remote indicating compass, is the function of two autotransformers. The 6-volt-ampere unit weighs 2 oz. and the 12-volt-ampere unit weighs 3 oz. Dimensions are approximately 1¾ x 1¼ x 1$\frac{1}{16}$ in., and both



operate over a frequency range from 380 to 420 cycles. These units can be applied where a 110-120 volt, a.c. power source is available. With sufficient ventilation the autotransformers will operate efficiently at any altitude. The core and coil units are impregnated with Glyptal clear varnish. All exposed solder joints are coated with red Glyptal to resist corrosion and the mounting clamps and terminals are also protected. While rated at 120 volts input, the units are suitable for operation over a voltage range from 110 to 130 volts. General Electric Co., Schnectady, N. Y.

Multi-Contact Timer

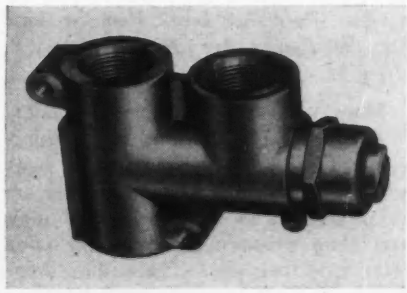
Automatic on-and-off sequence control of electrical circuits is provided by a multi-contact timer that consists essentially of a self-starting synchronous motor connected through a series of cut gears to a shaft on which adjustable phenolic-plastic cams are mounted. These cams operate a spring-contact mechanism that controls the load. The mechanism is mounted on a common base and can be enclosed. The starting mechanism is operated from a remote-



control momentary-start button so that the timer will go through its complete cycle or any part of it and stop. Reclosing the button will either continue or repeat the cycle. The timer is available for 110- and 250-volt, 60- and 50-cycle standard circuits and 25 cycles for some applications. Up to 18 contacts can be supplied but this varies according to specifications, as does the sequence of operation. Pure silver contacts are normally rated 10 amp. at 110 volts a.c. or 5 amp. at 220 volts a.c. non-inductive load. Housing dimensions are 5 in. wide, 4½ in. high and 8 to 15⅝ in. length for units having from 1 to 15 circuits. R. W. Cramer Co., Centerbrook, Conn.

Hydraulic Relief Valve

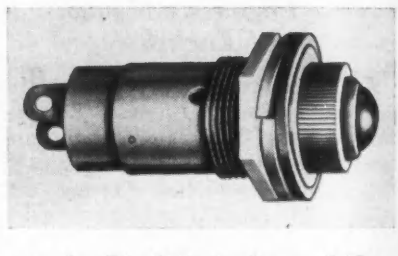
Spring loading with external regulation to control pressure regulation features a hydraulic pressure-relief valve made principally of aluminum. It is designed for chatterless operation and minimum pressure fluctuation. Pressure range is 1,000 to 2,100 lb. per sq. in. Various size models available in accordance with AN specifications pro-



vide capacities from 1.2 to 6 gal. per min. They operate through a temperature range from —65 deg. F. to 160 deg. F. and are suitable for all common aviation hydraulic fluids. Pesco Products Co., 11610 Euclid Ave., Cleveland, Ohio.

Pilot Light

Known as Series No. 1110, a pilot light is primarily for ungrounded panels. All variations are equipped with two solder terminals. Models No. 1110 (faceted jewel) and No. 1111 (plain jewel) take long bulbs. Models No. 1112 (faceted jewel) and No. 1113 (plain jewel) take round bulbs. These units are available with bayonet sock-



ets only. The light is also available as a shutter type for applications that require variable intensities. Models No. 1114 (faceted jewel) and No. 1115 (plain jewel) provide for round bulbs. They can also be furnished with polarized lens. All models mount in an ½-in. hole, having ½-in. jewels and are removed from the front of the panel. Gothard Mfg. Co., Springfield, Ill.

Special Electron Tubes

From high-frequency power for radio broadcasting and industrial heating applications to the measurement of low light intensities of the order of starlight is the range covered by four recently announced special electron tubes. RCA-9C22, a forced-air-cooled triode 25 in. high and weighing 235 lb., and its sister, the water-cooled RCA-9C21, are both suited for use in the class B modulator stage and in the plate-modulated class C final amplifier stage of high-power transmitters, as well as in large

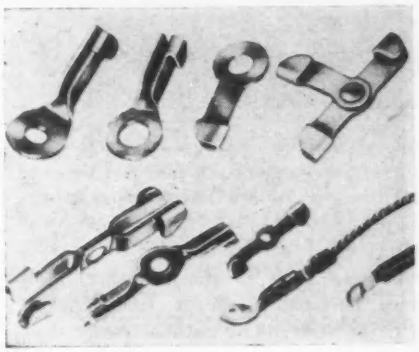
units for high-frequency heating. They can be used with maximum ratings at frequencies as high as 5 megacycles and with reduced ratings at frequencies up to 25 megacycles. Design features include a multi-strand, single-phase tungsten filament and an entrant metal header which provides short internal connections between the filament and filament terminal. RCA-2D21, illustrated, a thryatron 1⅞ in. high and weighing 0.5 oz. provides stable operations and a high control ratio for a variety of functions, such as an electronic switch. It will carry 0.1 amp. plate current continuously, and for periods up to 6 sec. out of 30 will carry 0.5 amp. It is filled with xenon. Other features include quick heating, wide temperature range, low internal drop, low preconduction current, versatility of control and high sensitivity. RCA-931-A is a 9-stage multiplier phototube capable of amplifying signals up to more than 200,000 times. It has been used in astronomical studies to measure the intensity of starlight. This sensitivity is made possible by the use of secondary emission as cathode electrons are impelled against nine successive dynodes before they reach the plate. Operated at 100 volts per stage, the tube has a sensitivity of 2 amp. per lumen. It does not need high-gain amplifier stages. Typical application of the 931-A is in quantitative spectrographic analysis. RCA Victor Div., Radio Corporation of America, Camden, N. J.

Metal Protective Film

Anti-corrosion films for metal are provided by Sublan, a heavy paste substitute for lanolin that has the same qualities. The coating passed tests against hydrobromic acid and salt-water immersion and humidity conducted in accordance with Government Specifications AXS 674. The film, which suppresses latent fingerprints on polished steel surfaces, is removed by cold solvent wash or dip. Glyco Products Co., 26 Court St., Brooklyn 2, N. Y.

Solderless Wire Connectors

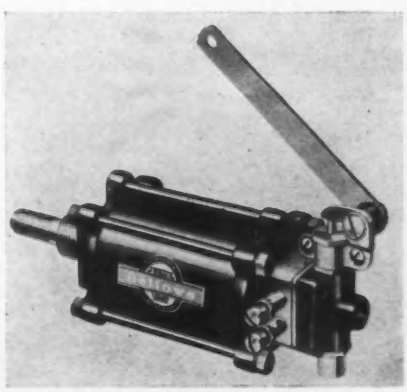
Based on a splicing terminal in which identical ends are put into 4-point electrical connection by knife-wiping action, a system of solderless knife-disconnect splicing has been developed. The splice has been adapted to "T-Link,"



"Y-Link," "H-Link" and "Cross-Link" applications, and to stub tabs, jumpers and small electrical assemblies such as switches, relays and other applications. Aircraft-Marine Products, Inc., 1591H N. Fourth St., Harrisburg, Pa.

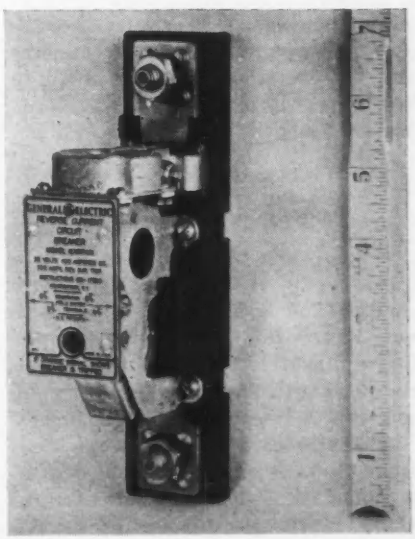
Air Motor

Known as Model BM, a new air motor is an improved version of the Bellows-Senacon air motor. The new unit makes automatic such operations as feeding work to tools, tools to work, opening and closing vises and holding fixtures, operating clutches and latches, and provides a source of flexible power for various stacking, clamping and embossing operations. The motor is a self-contained unit with integral valve, operating level and speed controls. The valve operating lever is adjustable to any angle in any plane and is synchronized by linkage, cam or solenoid controls to the machine cycle for automatic operation. Dual-exhaust throttle controls regulate speed of the piston rod in either or both directions. The air exhaust stream can be used to eject parts from holding devices or to clean chips and dirt away from the work area. The motor operates on any air pressure up to 175 lb. per sq. in. Power thrust is approximately five times operating air-line pressure. It is available in strokes ranging from 1½ in. up to 48 in. Heads are steel forgings. All moving parts are made of precision machined steel. Special piston construction reduces friction loss. Nose and foot mounts are standard equipment, with front-shelf and pivot mountings also available. Bellows Co., Akron, Ohio.



Air Circuit Breaker

Designated Type XRP, a new d.c. reverse-current air circuit breaker is designed especially for aircraft use. Rated 400 amp., 30 volts, with a reverse-current trip sitting of 300 amp. and interrupting rating of 8,000 amp. up to 50,000 ft. altitude, the breaker is connected in the generator circuit between the generator relay and the positive bus —one being used in the circuit of each generator. The current-interrupting capacity is sufficient for systems employing up to six 300-amp. generators and up to three storage batteries. When the generator speed is reduced to a point at which the voltage generated is lower than the bus voltage, the generator relay

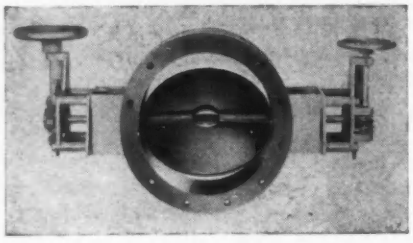


disconnects the generator from the bus. The breaker does not trip because the reverse current is lower than its trip setting. However, when a fault occurs in the generator circuit and the reverse current rises quickly to exceed the ca-

pacity of the generator relay, the air circuit breaker trips, interrupting fault currents in less than 0.015 sec. It will operate in the temperature range from —65 deg. F. to 160 deg. F. and will withstand 95 percent humidity at 160 deg. F. on 48-hr. test. In addition, it will meet the 200-hr. salt-spray test. It is not affected by vibrations from 5 to 55 cycles per sec. at $\frac{3}{32}$-in. amplitude applied in any direction. The 20-oz. breaker will not trip when closed and will not close contacts from the open position when subjected to shock or a linear acceleration of 30G. in any direction. General Electric Co., Schenectady, N. Y.

Two-In-One Valve

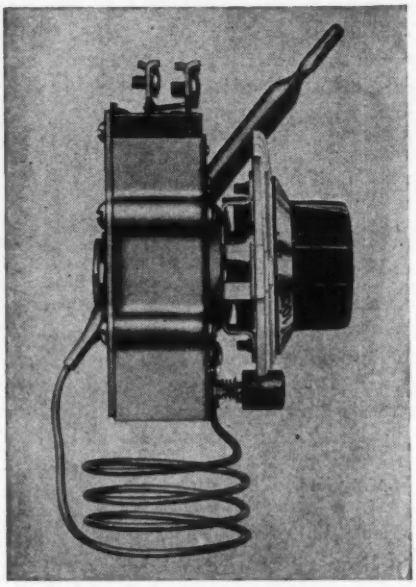
Designed for either a high-pressure drop and small volume or a low-pressure drop and large volume, Type 608 butterfly valve operates beyond the limitations of a single valve. Close control and shut-off of volume and pressure can be obtained and the valve is adaptable to power operation. The larger beveled vane seats against the body of the valve while the smaller vane is free revolving. Four to six



revolutions of the hand wheel open or close either vane. This valve is available in various combinations of sizes and for operating pressures of 15 to 900 lb. per sq. in. R-S Products Corp., Wayne Junction, Philadelphia 44, Pa.

Temperature Control

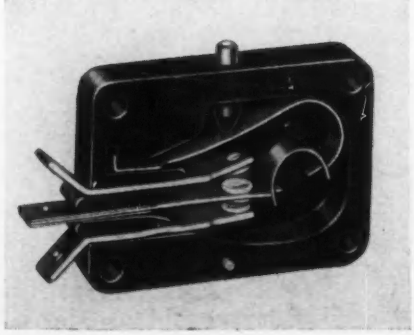
Manual reset overheating cut-off is a feature of the H 1 electric temperature control. The device consists of a standard Model D 1 double-pole, single-throw electric temperature control, equipped with an auxiliary single-pole, single-throw switch that automatically cuts off all current as soon as operating temperature is exceeded by 30 deg. F. in liquid or 50 deg. F. in air. During normal operation this control breaks both sides of the line. In case of excessive temperatures, the auxiliary switch is broken on one line only, which is sufficient to break the heating element circuit. The circuit is made by depressing the reset button. The contact is held in place by a locking device, or



safety catch, which is not disturbed by normal operation and does not operate until excess temperature expands the diaphragm. This movement raises the safety catch and causes the contacts to separate, even if the main contacts have become fused. A limit control without the temperature controlling mechanism is also available. Commercial & Industrial Div., Robertshaw Thermostat Co., 30 Church St., New York 7, N. Y.

Small Snap-Switch

Dimensions of $\frac{17}{64}$ in. thickness, $\frac{13}{16}$ in. height and $1\frac{3}{8}$ in. length feature the Miniac snap switch that operates on the rolling-spring principle. Fully enclosed in a phenolic-plastic case with four mounting holes of $\frac{3}{32}$ in. dia. actuation is with a stainless-steel pin plunger. All parts are corrosion resistant and contacts are of fine silver. Main blade, contact blade and rolling-spring are made of beryllium copper. Rating is 15 amp., 115 volts a.c. The switch is furnished in single-pole, normally open and normally closed, double-throw arrangements. It is also designed to permit leaf-type or over-travel plunger-type actuators to be attached to the case. Acro Electric Co., 1310 Superior Ave., Cleveland 14, Ohio.

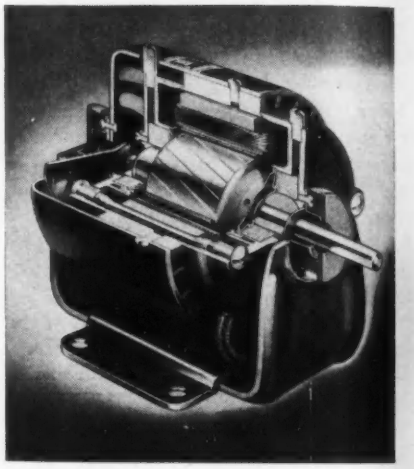


Notched Blind Rivet

Addition of a notch in the pulling mandrel of the self-plugging type Cherry rivet serves to inhibit the flow of the metal in the upsetting process when the head is formed. This makes for greater uniformity in the rivets. The notch also saves material because a shorter mandrel is required, which allows shorter drawbolt movement. Cherry Rivet Co., Los Angeles 13, Calif.

Miniature Motors

Power requirements from 1/125 to 1/20 hp. are handled by Type "T" Shaded-Pole Micromotors. Base-mounted models incorporate "Pressure-Locked" rubber mountings, consisting of two large rubber pads, two spring-steel lock rings and a base welded into a single, rigid unit. Flush-Weld rotors, in which slots are copper filled flush with circumference and preformed copper inductors are fused permanently for maximum



conductivity, are used. Different performance characteristics can be met by modifying inductor area. Proper operation under extreme temperatures is assured by using inclosed cooling fins and fan. Frames are of three-piece die-cast design machined for uniformity. Porous bronze bearings, spring-tension centered, filter the lubricant. Stator windings are impregnated with special varnish. Special voltages, frequencies, leads, finishes and shaft extensions are available. A. G. Redmond Co., Owosso, Mich.

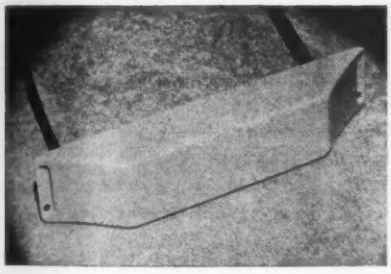
Dial Thermometer

Especially suited to installations requiring a long length of tubing between the bulb and the instrument, a 4½-in. dial thermometer has a phenolic case. This mercury-actuated instrument is built to withstand severe operating

conditions and is available with Accuratus tubing. The plastic case is resistant to corrosion and shock. The metal dial is finished in black with white figures. The 53K series thermometer can be had with a wide variety of standard temperature-sensitive bulb constructions and is available in the following ranges: —40 deg. F. to 120 deg. F., 40 deg. F. to 220 deg. F., 350 deg. F. to 550 deg. F., 400 deg. F. to 900 deg. F., —10 deg. F. to 110 deg. F., 80 deg. F. to 300 deg. F., 300 deg. F. to 750 deg. F., 30 deg. F. to 150 deg. F., 50 deg. F. to 400 deg. F. and 200 deg. F. to 900 deg. F. Taylor Instrument Companies, P. O. Box 110, Rochester 1, N. Y.

Streamlined Ballast

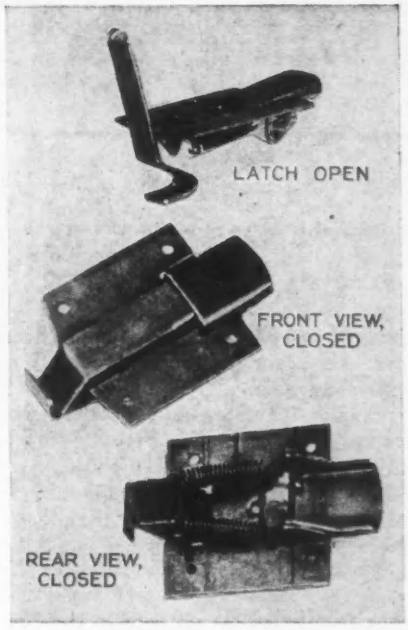
To fit the contours of lightweight fluorescent fixtures designed to meet waritme limitations on metal, Tulamp 40-watt ballasts are housed in streamlined drawn-steel cases. The units are arranged for external mounting on top of the fixture with leads from the bottom of the case. This makes possible installation on a narrow and shallow wiring channel in an exposed position and allows appreciable reduction in the amount of critical metal required for the manufacture of the fixture. The ballasts operate in lower ambient temperatures, which contribute to long



life. The case has a dark-gray finish. The units comply with local regulations concerning power factor and do not differ electrically from conventional ballasts of corresponding ratings. General Electric Co., Schenectady, N. Y.

Quick-Access Fastener

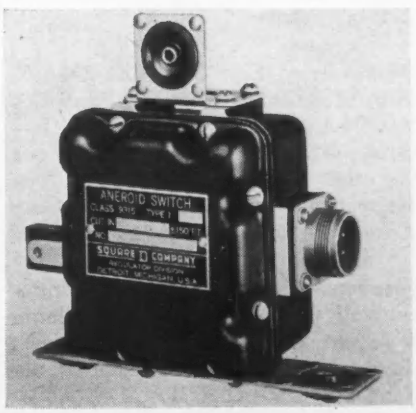
To replace non-flush fasteners in applications that require frequent opening and closing such as inspection and access doors, the M-Loc fastener features speedy installation. Designed specifically for aircraft, it lies flush, creating no air resistance. No mating latch parts are required and no tools are needed for operation. Finger pressure on the plate of the latch opens the door and pressure on the door locks the latch. Weight is saved because of the



sheet-metal construction, no castings being used. An exclusive feature is the angle in the tongue, permitting the tongue to by-pass a reinforcing flange around the opening. Simmonds Aerocessories, Inc., 21-10 49th Ave., Long Island City, N. Y.

Aneroid Switch

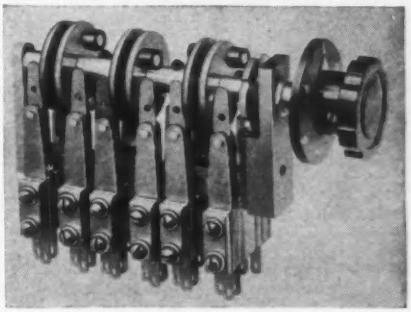
For close control of applications involving electrical control responsive to altitude change, the Class 9315 aneroid switch is comprised of a precision-switch mechanism actuated by a sensitive diaphragm or bellows of the evacuated and temperature-compensated type. The mechanism is mounted in a die-cast anodized aluminum housing provided with a gasketed cover. The manifold or atmospheric pressure connection is through a ⅛-in. tapped hole. The unit is provided with an approved two-prong, pressure-proof fitting for plug-in electrical connection to the switch. Three-point mounting minimizes effects of vibration and shock. The switch can be used over a temperature range of —65 deg. F. to 160 deg. F. The operating



range is up to 35,000 ft. with a minimum differential of 1,000 ft. Net weight is 24 oz. At 28 volts d.c. the switch will handle up to 25 amp. inrush current. Regulator Div., Square D Co., 6060 Rivard St., Detroit, Mich.

Rotary Cam Lever Switch

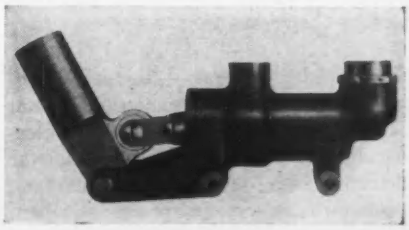
Designed for one to six index positions is a rotary cam lever switch. Any combination of spring-leaf contact assemblies is available for each position. It is adaptable to opening or closing numerous circuits, in sequence or repeat, with a single control knob. Contact assemblies in any section can be lifted from the frame by removing a single bolt. Features include circular cams for locating up to twelve spring-type actuators, cast aluminum frame, phenolic plastic cams and rollers, static shielding, nickel-plated phosphor bronze contact springs, solid silver contacts and



contact build-ups assembled under pressure. The switch is rated at 10 amp., 125 volts a.c. General Control Co., 1200 Soldiers Field Rd., Boston 34, Mass.

Hydraulic Pump

For installations in which small quantities of fluid at high pressures are required, such as in aircraft hydraulic systems, a hand-operated pump is offered. Displacement is 1.50 in. maximum per complete cycle. Rated capacity is given as 0.125 gal. per min. at 20 cycles per min. with 1,500 lb. per sq. in. discharge pressure. Lever arrangement consists of a fixed handle with 57.5 deg. total movement. Needle bearings at lever and pivot minimize operating torque. Weight is 2.4 lb. Pesco Products Co., 11610 Euclid Ave., Cleveland 6, Ohio.



Manufacturers' Publications

Ball Bearings—Marlin-Rockwell Corp., Jamestown, N. Y. Bulletin Protfolio. Contains 27 bulletins covering all phases of ball bearings including housings, covers, seals, lubricants, mountings, preloading, fits and interchangeability tables among the information of interest to engineers.

Magnesium—Dow Chemical Co., Midland, Mich. Booklet, 76 pages. Gives statements filed by Dr. Willard H. Dow, president and general manager of the company, with the Truman committee and extracts from the cross-examination and the report of the committee, to explain the company's position in the development of magnesium.

Temperature Measurement—Leeds & Northrup Co., 4934 Stenton Ave., Philadelphia 44, Pa. Catalog N-33B, 48 pages. Brings up-to-date the company's 1941 catalog covering Rayotubes and Micromax equipment and describes equipment being used in new applications.

Grommet—Arens Controls, Inc., 2253 S. Halstead St., Chicago 8, Ill. Bulletin, 4 pages. Describes Dura-Grom, a fume- and water-tight vibration-resistant grommet for supporting lines through bulkheads and firewalls.

Synthetic Rubber—Acadia Synthetic Products Division, Western Felt Works, 4115 Ogden Ave., Chicago, Ill. Booklet, 70 pages. Contains much information and data about types of synthetic rubber, processed by the company, to meet a wide variety of applications.

Connectors—Cannon Electric Development Co., 3209 Humboldt St., Los Angeles 31, Calif. Bulletin, 12 pages. Photographs, dimensional drawings, data and other information about laboratory and switchboard connectors are presented.

Drawing Instruments—Charles Bruning Co., 100 Reade St., New York 13, N. Y. Bulletin, 4 pages. Describes features of drawing instrument sets handled by the company.

Electrical Instruments—Norton Electrical Instrument Co., Manchester, Conn. Catalog, 32 pages. Moving-coil, electromagnetic and electrodynamometer instruments in the form of ammeters, voltmeters and wattmeters are described.

Industrial Glassware and Accessories—Industrial Div., Corning Glass Works, Corning, N. Y. Bulletins Nos. 838. 4 pages; 840, 4 pages; 841 2 pages; 843, 4 pages; and 844, 8 pages. Bulletin 838 describes Vycor, 96 percent silica, industrial glassware, including application and design data; bulletin 840 shows adaptor connections for Pyrex piping; bulletin 841 describes glass valves for Pyrex piping; bulletin 843 gives size data on Pyrex tubing; and bulletin 844 covers the properties of Pyrex No. 774 glass.

Electric Heaters—H. & A. Mfg. Co., 86 Leroy Ave., Buffalo 14, N. Y. Bulletin, 16 pages. Development of electric contact heaters is described; sizes, shapes and capacities available are indicated; and applications are given.

Reproduction—Charles Bruning Co., 100 Reade St., New York 13, N. Y. Bulletin 4 pages. Describes features and limitations of the Copyflex 55C continuous printer designed especially for photographic materials.

Aluminum—Aluminum Div., Reynolds Metals Co., 2500 S. Third St., Louisville, Ky. Bulletin 22-A, 8 pages. Presents engineering and design data covering aluminum sheet and strip.

Metal Tubing—American Metal Hose Branch, American Brass Co., Waterbury, Conn. Bulletin SS50, 40 pages. Covers manufacture, uses, types available, physical properties and installation methods of seamless flexible metal tubing.

Wire—Callite Tungsten Corp., Union City, N. J. Data Chart. Various data of value to users of tungsten and molybdenum wire and rod are given on this slide-type chart.

Oilers—Trico Fuse Mfg. Co., 2948 N. Fifth St., Milwaukee 12, Wis. Bulletins Nos. 24-A and 25-A, 2 pages each. Two variations, one using glass bottles and the other using plastic bottles, of an automatically controlled visible oiling method are described.

Forgings—Titusville Forge Div., Struthers Wells Corp., Titusville, Pa. Bulletin, 4 pages. Illustrates typical forgings, shafting and crankshafts made by the company.

Packaging — Angier Corp., Framingham, Mass. Bulletin, 6 pages. Describes a method of wrapping engines and parts in paper so that they can be delivered to battlefronts unaffected by corrosive influences.

Insulation Testing—Herman H. Sticht Co., 27 Park Place, New York 7, N. Y. Bulletin No. 445, 8 pages. Design, features and operation of the C-2 Megohmer insulation tester, operated by a rechargeable storage battery, are described.

Research—American Council of Commercial Laboratories, 63 Wall St., New York 5, N. Y. Lists independent testing, research and inspection laboratories and gives executive personnel, customary services and specialties of each organization.

Transmissions—Winfield H. Smith, Inc., Springville, Erie County, N. Y. Brochure, 24 pages. Typical applications and installations of speed reducers, gears and transmission machinery made by the company are shown.

Plastic Pipe—Hodgman Rubber Co., Framingham, Mass. Folder No. P-10, 4 pages. Saran pipe and fittings are described and sizes, weights and bursting pressures are given.

Casters—Bassick Co., Bridgeport 2, Conn. Catalog No. 122, 96 pages. Features and descriptive data on numerous types of truck casters are given, prefaced by information on selection.

Tensile Testing—W. C. Dillon & Co., 5410 W. Harrison St., Chicago 44, Ill. Bulletin No. 141, 8 pages. Features of Model K portable tensile tester are covered.

Springs—Muehlausen Spring Corp., 1943 Michigan Ave., Logansport, Ind. Folder. Illustrates various types of springs in connection with description of how large hot-coiled springs are made.

Coolant Pumps—Worthington Pump & Machinery Corp., Harrison, N. J. Bulletin W-321-B16, 4 pages. Details of vertical and circulating Monobloc centrifugal pumps are illustrated.

Electronics — Fansteel Metallurgical Corp., North Chicago, Ill. Booklet, 16 pages. Basis and uses of electronics are described briefly, then emphasis is placed on the use of electronics by the company for separating tantalum, molybdenum and tungsten from their ores in this article by F. L. Hunter, chief engineer of the electronics division.

Wrench Torque Nomogram for Aircraft Nuts

R. R. WIESE
Kaiser Cargo, Inc., Fleetwings Division

TABLE gives ultimate tensile strength, ultimate strength in single shear, and maximum bearing values for aircraft AN steel bolts used to fasten two or more sheets in a tension joint or connection, in which the bolts are subjected to shearing loads at sections lying in the plane of the sheet junctures and also bearing loads on the bolt body in contact with the sheets.

Tensile strength given for bolts is based on area for diameter at root of thread and an ultimate tension stress of 125,000 lb. per sq. inch.

Yield strength is calculated at 80 percent of the tensile strength of bolt.

Shear strength is based on area of unthreaded portion of bolt shank for a shearing stress of 75,000 lb. per sq. in. All values for bearing strength of sheet are based on full bolt diameter and total width of a single sheet. Sheets are assumed to have had a heat-treatment which gives a tensile strength of 125,000 lb. per sq. in., with a bearing strength of 175,000 lb. per sq. inch.

To increase the range of usefulness of the data, factors are also given for determining the bearing values of sheet having other tensile strengths.

PROPERTIES OF AIRCRAFT AN STEEL BOLTS IN TENSION JOINTS

Bolt size dia. and threads	10–32	1/4–28	5/16–24	3/8–24	7/16–20	1/2–20	9/16–18	5/8–18	3/4–16	7/8–14	1–14
Area (root dia.), sq. in.	0.0175	0.0326	0.0524	0.0809	0.1090	0.1486	0.1888	0.2400	0.3513	0.4805	0.6464
Tens. Str. (root dia.), lb.	2,188	4,075	6,550	10,112	13,625	18,575	23,600	30,000	43,913	60,063	80,800
Yield Str. (root dia.), lb.	1,750	3,260	5,240	8,090	10,900	14,860	18,880	24,000	35,130	48,052	64,640
Area of shank, sq. in.	0.0284	0.0491	0.0767	0.1105	0.1503	0.1964	0.2485	0.3068	0.4418	0.6013	0.7854
Single shear, lb.	2,126	3,682	5,752	8,284	11,275	14,746	18,637	23,010	33,134	45,099	58,905

Sheet thickness, in.	Bearing Strength of Sheet in Pounds for Alloy Steel of 125,000 lb. per sq. in. tensile strength										
0.022	735	963	1,208	1,443	1,683	1,926	2,170	2,405	2,886	3,370	3,852
0.028	936	1,225	1,540	1,840	2,145	2,450	2,760	3,060	3,680	4,280	4,900
0.034	1,125	1,490	1,870	2,235	2,605	2,975	3,355	3,720	4,460	5,200	5,950
0.050	1,655	2,190	2,750	3,285	3,830	4,375	4,930	5,480	6,560	7,660	8,740
0.063	2,085	2,760	3,470	4,140	4,830	5,520	6,210	6,880	8,280	9,660	11,040
0.078	2,580	3,420	4,300	5,120	5,970	6,840	7,700	8,540	10,250	11,940	13,640
0.094	3,110	4,130	5,180	6,170	7,200	8,250	9,280	10,280	12,350	14,400	16,440
0.109	3,610	4,780	6,000	7,150	8,350	9,560	10,750	11,920	14,310	16,700	19,100
0.125	4,140	5,480	6,880	8,200	9,580	10,950	12,320	13,680	16,420	19,140	21,880
5/32	5,170	6,850	8,580	10,240	11,960	13,680	15,400	17,080	20,520	23,900	27,360
3/16	6,200	8,210	10,280	12,228	14,320	16,380	18,450	20,460	24,600	28,630	32,800
1/4	8,280	10,980	13,720	16,400	19,140	21,880	24,650	27,400	32,900	38,300	43,800
5/16	10,350	13,750	17,200	20,550	23,960	27,400	30,860	34,300	41,190	47,950	54,840
Bolt size dia., in.	10	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1

Bearing values below heavy line are greater than maximum allowable single shear.

For bearing values other than 125,000 lb. per sq. in. tensile strength multiply value from table by proper K from

	Tensile Strength of Sheet, lb. per sq. in.		Corresponding bearing strength, lb. per sq. in.	K
Steel	90,000		140,000	0.800
Steel	150,000		190,000	1.085
Steel	180,000		200,000	1.141
Aluminum				
(14ST) $e/D = 1.5$	60,000		90,000	0.5143
(14ST) $e/D = 2.0$	60,000		114,000	0.6514
(17ST) $e/D = 1.5$	55,000		83,000	0.4743
(17ST) $e/D = 2.0$	55,000		105,000	0.6000
Bare (24ST) $e/D = 1.5$	62,000		93,000	0.5314
Bare (24ST) $e/D = 2.0$	62,000		118,000	0.6743
Clad (24ST) $e/D = 1.5$	56,000	Heat treated	84,000	0.4800
Clad (24ST) $e/D = 2.0$	56,000	by user	106,000	0.6057
Clad (24ST) $e/D = 1.5$	60,000		90,000	0.5143
Clad (24ST) $e/D = 2.0$	60,000		114,000	0.6514
Bare (24ST) $e/D = 1.5$	66,000		99,000	0.5657
Bare (24ST) $e/D = 2.0$	66,000		125,000	0.7143
Clad (24ST) $e/D = 1.5$	61,000	As furnished	91,000	0.5200
Clad (24ST) $e/D = 2.0$	61,000	by mill	116,000	0.6629
Clad (24ST) $e/D = 1.5$	64,000		96,000	0.5486
Clad (24ST) $e/D = 2.0$	64,000		122,000	0.6971

Remarks:
Figures based on values given in ANC-5 Amend. 1 10/22/43.
D = hole diameter; e = edge distance measured from hole centerline in direction of stressing. Use value of $e/D = 2.0$ for all larger values of edge distance.

Wrench Torque Nomogram for Aircraft Nuts

R. R. WIESE
Kaiser Cargo, Inc., Fleetwings Division

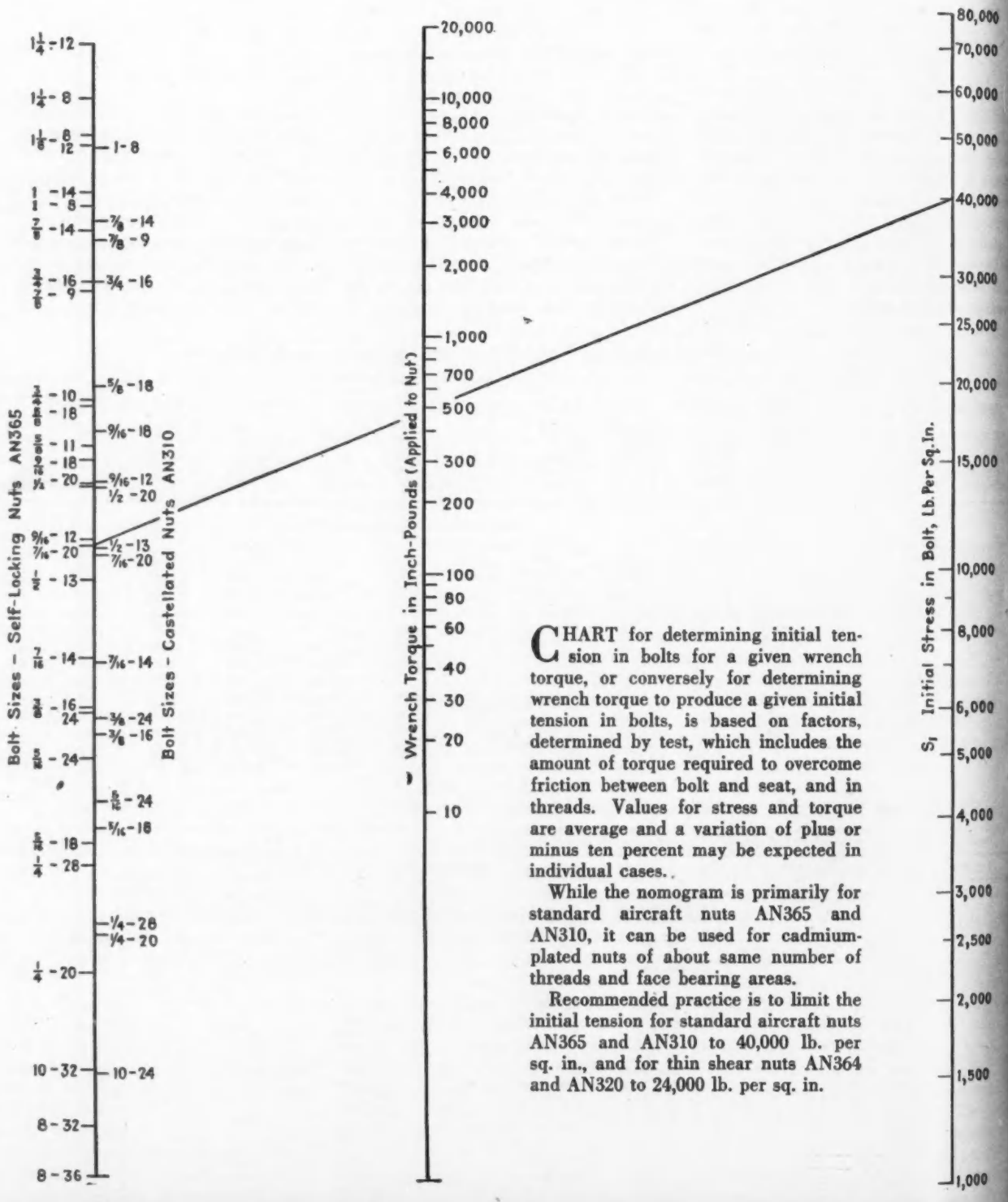


CHART for determining initial tension in bolts for a given wrench torque, or conversely for determining wrench torque to produce a given initial tension in bolts, is based on factors, determined by test, which includes the amount of torque required to overcome friction between bolt and seat, and in threads. Values for stress and torque are average and a variation of plus or minus ten percent may be expected in individual cases.

While the nomogram is primarily for standard aircraft nuts AN365 and AN310, it can be used for cadmium-plated nuts of about same number of threads and face bearing areas.

Recommended practice is to limit the initial tension for standard aircraft nuts AN365 and AN310 to 40,000 lb. per sq. in., and for thin shear nuts AN364 and AN320 to 24,000 lb. per sq. in.

PRODUCT ENGINEERING — REFERENCE BOOK SHEET